CONFORM AND RETURN

FRANKLIN BROCKWAY GOWDY, State Bar No. 47918
THOMAS D. KOHLER, State Bar No. 207917
MORGAN, LEWIS & BOCKIUS LLP
One Market, Spear Street Tower
San Francisco, CA 94105-1126
Tel: 415.442.1000
Fax: 415.442.1001

MICHAEL J. LYONS, State Bar No. 202284
MORGAN, LEWIS & BOCKIUS LLP
2 Palo Alto Square
3000 El Camino Real, Suite 700
Palo Alto, CA 94306-2212
Tel: 650.843.4000
Fax: 650.843.4001
mlyons@morganlewis.com

Attorneys for Plaintiff
NIDEC CORPORATION

ORIGINAL FILED
FEB 15 2005
Richard W. Wieking
Clerk, U.S. District Court
Northern District of California
San Jose

E-filing

UNITED STATES DISTRICT COURT

NORTHERN DISTRICT OF CALIFORNIA

NIDEC CORPORATION, Plaintiff, vs. VICTOR COMPANY OF JAPAN, LTD., JVC COMPONENTS (THAILAND) CO., LTD., KABUSHIKI KAISHA AGILIS, and AGILIS TECHNOLOGY INC., Defendants.	Case No. C05 00686 WDB **COMPLAINT FOR PATENT INFRINGEMENT** **DEMAND FOR JURY TRIAL**

Plaintiff Nidec Corporation ("Nidec" or "Plaintiff"), as and for its claims for relief herein, alleges on information and belief as follows:

GENERAL ALLEGATIONS AND PARTIES

1. In this action, Nidec seeks damages and injunctive relief against Victor Company of Japan, Ltd. ("JVC"), JVC Components (Thailand) Co., Ltd. ("JVC Thailand"), Kabushiki Kaisha Agilis ("KK Agilis") and Agilis Technology Inc. ("Agilis"). JVC, JVC Thailand, KK Agilis and Agilis are referred to collectively herein as "Defendants."

2. Nidec is a foreign company having a principal place of business at 338 Tonoshiro-cho, Kuze Minami-ku, Kyoto 601-8205, Japan.

3. JVC is a foreign company having a principal place of business at 12, Moriya-cho 3-chome, Kanagawa-ku, Yokohama, 221- 8528, Japan.

4. JVC Thailand is a foreign company having a principal place of business at Suranaree Industrial Zone 555, Moo 6, Tambol Nong Rawieng, Amphur Muang, Nakhonratchasima, 3000, Thailand.

5. KK Agilis is a foreign company having a principal place of business at Shinjuku West Building 3rd Floor, 8-2-5, Nishi-Shinjuku, Shinjuku, Tokyo 160-0023, Japan.

6. Agilis is a corporation organized and existing under the laws of the State of California, having a principal place of business and having only a single location at 568 East Weddell Drive, Suite 7, Sunnyvale, California 94089.

7. On information and belief, Nidec alleges that JVC and JVC Thailand manufacture spindle motors, and directly and/or through their intermediaries KK Agilis and Agilis have and are importing, offering for sale and selling these spindle motors in the Northern District of California.

JURISDICTION AND VENUE

8. Nidec brings this action pursuant to the Patent Laws of the United States, Title 35, United States Code.

9. Jurisdiction is conferred on this Court pursuant to Title 28, United States Code, sections 1331 (federal question jurisdiction) and 1338(a) (original jurisdiction under patent laws).

10. Venue is proper in this judicial district under the provisions of Title 28, United States Code, sections 1391(b) and (c) (general venue statute) and 1400(b) (civil action for patent infringement).

PATENT INFRINGEMENT

11. Nidec is the owner of the entire right, title and interest in and to United States Patent No. 5,667,309 ("the '309 patent"), United States Patent No. 6,343,877 B1 ("the '877 patent"), United States Patent No. 6,554,476 B2 ("the '476 patent"), and United States Patent No.

6,793,394 B2 ("the '394 patent"). The '309 patent, the '877 patent, the '476 patent, and the '394 patent are referred to collectively herein as the "Patents-In-Suit." A true copy of each of the Patents-In-Suit are attached hereto as Exhibits A through D. Each of these patents discloses and claims subject matter generally relating to spindle motors.

12. Nidec alleges on information and belief, that the Defendants have infringed and continue to infringe the Patents-In-Suit by making, using, selling, offering for sale and/or by importing into the United States devices that embody or otherwise practice one or more of the claims of the Patents-In-Suit, or by otherwise contributing to infringement or inducing others to infringe the Patents-In-Suit. These acts constitute violations of Title 35, United States Code, section 271.

13. Nidec has given notice to JVC that the JVC spindle motors and/or devices used in, for example, hard disk drives, embody or otherwise practice the claimed subject matter of the Patents-In-Suit. Defendants' infringement of the Patents-In-Suit is, has been, and continues to be willful and deliberate.

14. Unless enjoined by this Court, the Defendants will continue their acts of infringement causing substantial and irreparable harm to Nidec.

15. As a direct and proximate result of Defendants' infringement of the Patents-In-Suit, Nidec has been and continues to be damaged in an amount yet to be determined.

16. This is an exceptional case within the meaning of Title 35, United States Code, section 285, and Nidec is accordingly entitled to an award of its attorneys' fees.

17. WHEREFORE, Nidec demands judgment:

A. Preliminarily and permanently enjoining and restraining the Defendants, their officers, directors, employees, agents, servants, successors and assigns, and any and all persons acting in privity or in concert with the Defendants, from further infringement of the Patents-In-Suit;

B. Assessing against Defendants and awarding to Nidec damages sufficient to compensate for Defendants' infringement of the Patents-In-Suit, and conducting an accounting to determine said damages, as provided by Title 35, United States Code, section 284;

C. Increasing said damages to three times the amount found or assessed;

D. Awarding Nidec its costs and disbursements in this action, including reasonable attorneys' fees, as provided by Title 35, United States Code, section 285; and

E. Granting to Nidec such other and further relief as this Court may deem just and proper.

DEMAND FOR JURY TRIAL

18. Plaintiff Nidec hereby requests a trial by jury.

CERTIFICATION OF INTERESTED ENTITIES OR PERSONS

19. Pursuant to Civil L.R. 3-16, the undersigned certifies that the following listed persons, associations of persons, firms, partnerships, corporations (including parent corporations) or other entities (i) have a financial interest in the subject matter in controversy or in a party to the proceeding, or (ii) have a non-financial interest in that subject matter or in a party that could be substantially affected by the outcome of this proceeding:

a. Mr. Shigenobu Nagamori, President and CEO of Nidec Corporation;

b. Japan Trustee Services Bank, Ltd.;

c. The Master Trust Bank of Japan, Ltd.; and

d. Matsushita Electric Industrial Co., Ltd. (stockholder of Victor Company of Japan, Ltd.).

///

///

///

///

///

///

///

///

Dated: February 15, 2005

Respectfully Submitted,

MORGAN, LEWIS & BOCKIUS LLP

FRANKLIN BROCKWAY GOWDY
THOMAS D. KOHLER
MICHAEL J. LYONS

By:

Michael J. Lyons
Attorneys for Plaintiff
NIDEC CORPORATION

CERTIFICATE OF SERVICE

I am employed in the City of Palo Alto, County of Santa Clara, State of California, I am over the age of 18 years and not a party to the within action. My business address is 2 Palo Alto Square, 3000 El Camino Real, Palo Alto, California 94306. On February 16, 2005, I caused copies of the attached document(s) described as follows:

COMPLAINT
SUMMONS IN A CIVIL SUIT
FRCP 7.1 DISCLOSURE
ORDER SETTING CASE MANAGEMENT CONFERENCE
STANDING ORDER OF JUDGE WAYNE D. BRAZIL
DECLINATION TO PROCEED BEFORE A MAGISTRATE JUDGE(FORM)
CONSENT TO PROCEED BEFORE A MAGISTRATE JUDGE (FORM)
NOTICE OF ASSIGNMENT OF CASE
GUIDELINES OF THE NORTHERN DISTRICT OF CALIFORNIA
WAIVER OF SERVICE OF SUMMONS (FORM)
NOTICE OF LAWSUIT (FORM)
GENERAL ORDER NO. 40
ADR PROCESS MATERIALS
ECF REGISTRATION INFORMATION HANDOUT
DISPUTE RESOLUTION PROCEDURES PAMPHLET

to be served on:

AGILIS TECHNOLOGY INC.
568 East Weddell Drive, Suite 7,
Sunnyvale, California 94089

______(BY OVERNIGHT DELIVERY) I caused each such envelope to the addressee(s) noted above, with charges fully prepaid, to be sent by overnight delivery from Palo Alto, California. I am readily familiar with the practice of Morgan, Lewis & Bockius LLP for collection and processing of correspondence for overnight delivery, said practice being that in the ordinary course of business, mail is placed with the overnight delivery service on the same day as it is placed for collection.

__BY ELECTRONIC MAIL) The person whose name is noted below caused to be transmitted by electronic mail each such document to the addressee(s) noted above.

__(BY FIRST CLASS MAIL) I caused each such envelope to the addressee(s) noted above, with postage thereon fully prepaid, to be placed in the United States mail in Palo Alto, California. I am readily familiar with the practice of Morgan, Lewis & Bockius LLP for collection and processing of correspondence for mailing, said practice being that in the ordinary course of business mail is deposited in the United States Postal Service the same date as it is placed for collection; and

______(BY FACSIMILE) The person whose name is noted below caused to be transmitted by facsimile each such document to the addressee(s) noted above; and

_X_(BY PERSONAL SERVICE) The person whose name is noted below caused to be delivered by hand each such envelope to the addressee(s) noted above.

I declare under penalty of perjury under the laws of the State of California that the foregoing is true and correct. Executed at Palo Alto, California, on February 16, 2005.

James H. Phan

EXHIBIT A
TO COMPLAINT

US005667309A

United States Patent [19]

Nose

[11] **Patent Number: 5,667,309**

[45] **Date of Patent: Sep. 16, 1997**

[54] **BEARING SEAL SYSTEM**

[75] Inventor: **Tamotsu Nose**, Nagano, Japan

[73] Assignee: **Sankyo Seiki Mfg. Co., Ltd.**, Nagano-ken, Japan

[21] Appl. No.: **557,845**

[22] Filed: **Nov. 14, 1995**

[30] **Foreign Application Priority Data**

Nov. 15, 1994	[JP]	Japan	6-305584
Nov. 29, 1994	[JP]	Japan	6-319194
Dec. 6, 1994	[JP]	Japan	6-330228
Dec. 9, 1994	[JP]	Japan	6-331748

[51] **Int. Cl.**[6] **F16C 32/06**; F16C 33/72

[52] **U.S. Cl.** **384/132**; 384/133; 277/80

[58] **Field of Search** 384/107, 114, 384/119, 124, 132, 133; 277/80

[56] **References Cited**

U.S. PATENT DOCUMENTS

4,883,367	11/1989	Maruyama	384/133 X
4,890,850	1/1990	Raj et al.	277/80
5,112,142	5/1992	Titcomb et al.	384/107
5,145,266	9/1992	Saneshige et al.	384/132 X
5,372,432	12/1994	Ishikawa	384/133
5,423,612	6/1995	Zang et al.	384/132 X
5,427,456	6/1995	Hensel	384/112

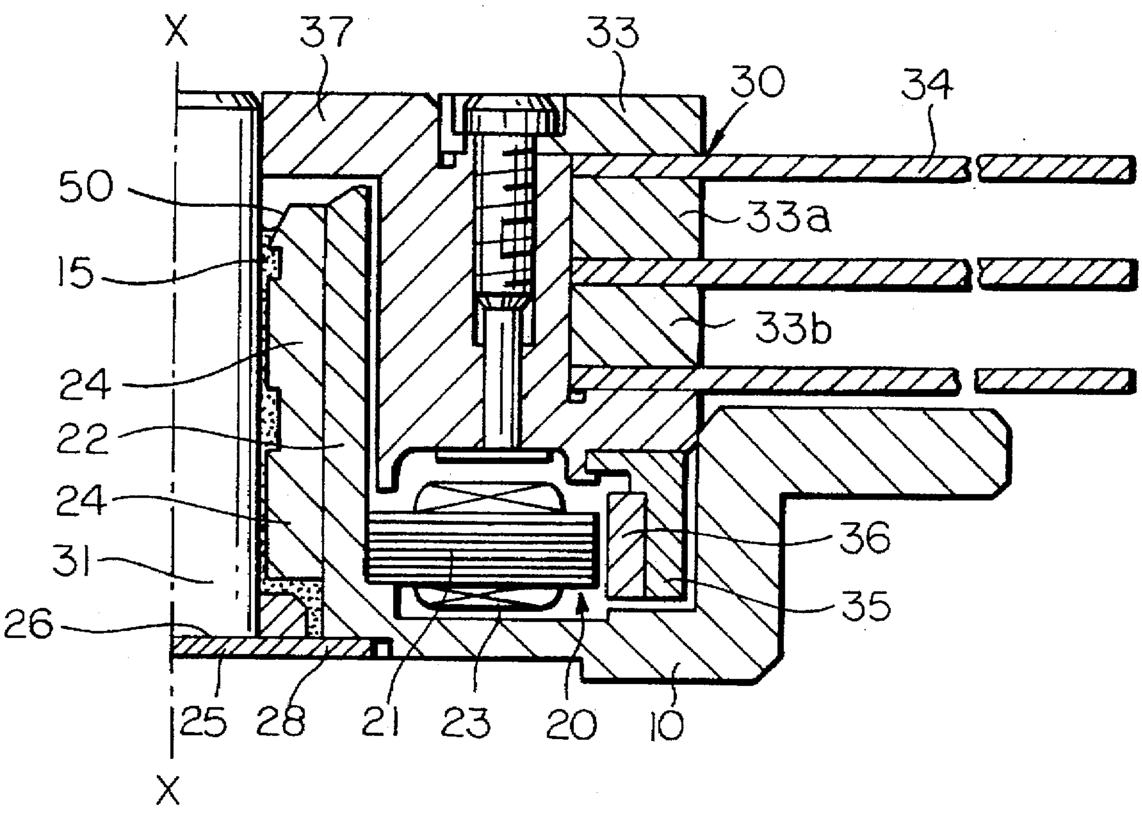
Primary Examiner—Thomas R. Hannon

Attorney, Agent, or Firm—McAulay Fisher Nissen Goldberg & Kiel, LLP

[57] **ABSTRACT**

A bearing seal system having a bag structure in a first embodiment wherein (1) oil is filled in a tapering bag space section positioned from the bag section to an exit section; (2) the space in the tapering space section is kept to 0.8 mm or less the slope angle of the outer end of the tapering space section is kept to 45° or less. In this way, oil is kept from air and a stable no-leak status is made possible; (3) the capacity of the tapering space section is made larger than that of a bag section or a radial bearing section in order to retain oil in the bearing section at all times and to prevent oil from leaking as a result of variations in the quantity of oil injected or in the capacity; and (4) the space ratio of the inner end and the outer end of the tapering space section is made larger to prevent oil from moving to the bearing section even when air migrates into the oil surface. In a second embodiment, the tapering space section where the surface of lubrication oil is injected can be at two places at both ends of a radial bearing section or at both ends of a bearing with a radial bearing section and a third bearing section. A third embodiment is disclosed.

21 Claims, 9 Drawing Sheets



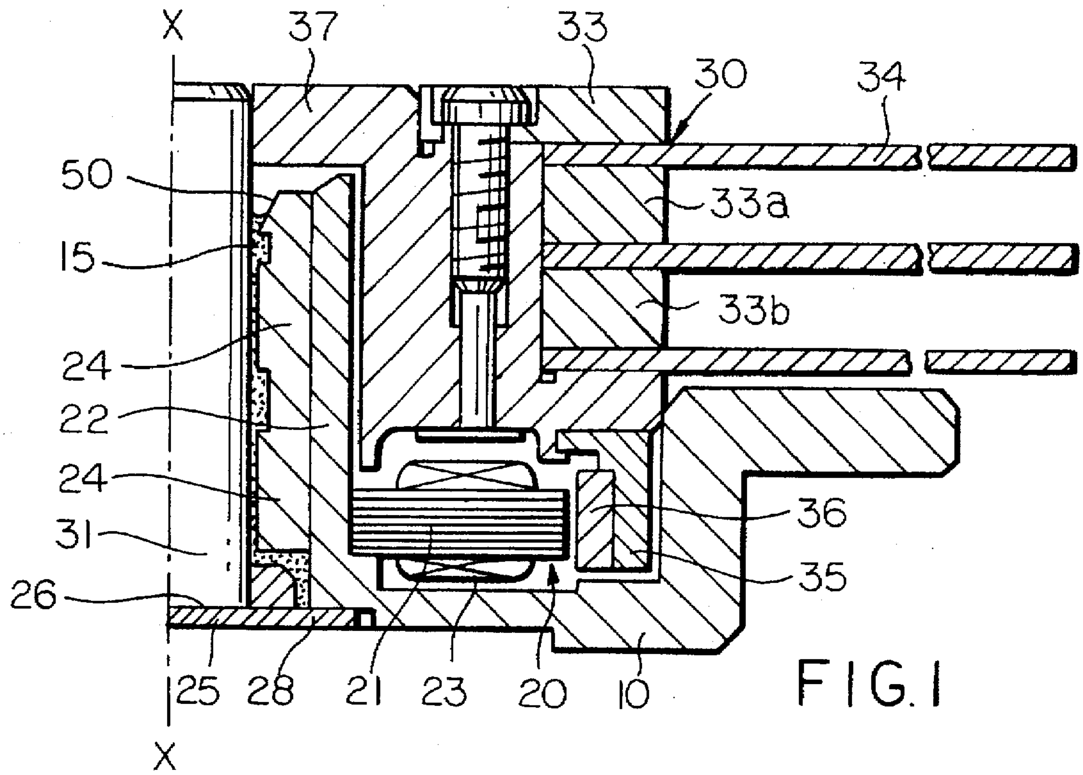

X
37
33
30
34
50
33a
15
33b
24
22
24
36
31
35
26
25
28
21
23
20
10
X

FIG.1

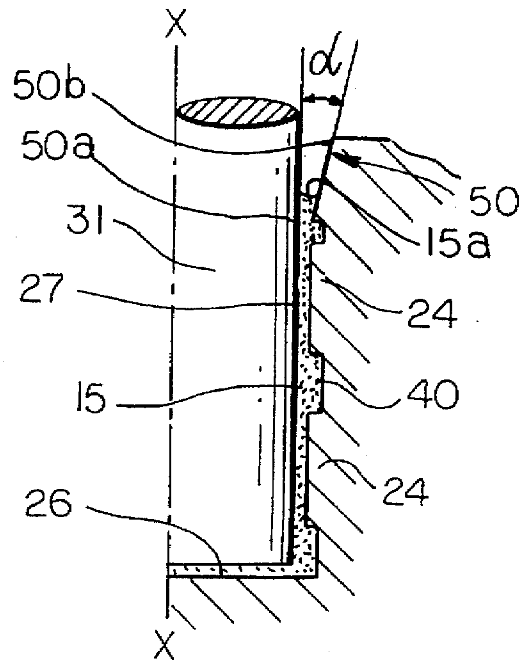

X
α
50b
50a
50
15a
31
27
24
15
40
26
24
X

FIG.2

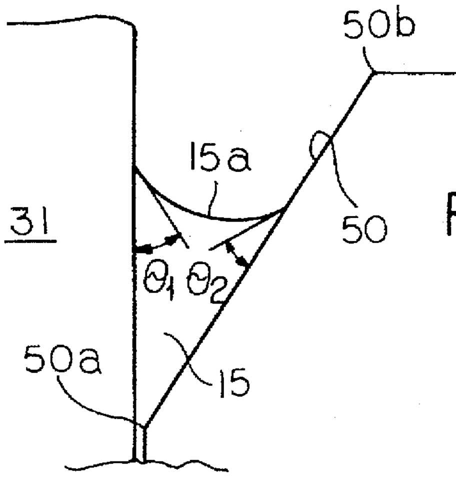


FIG. 3

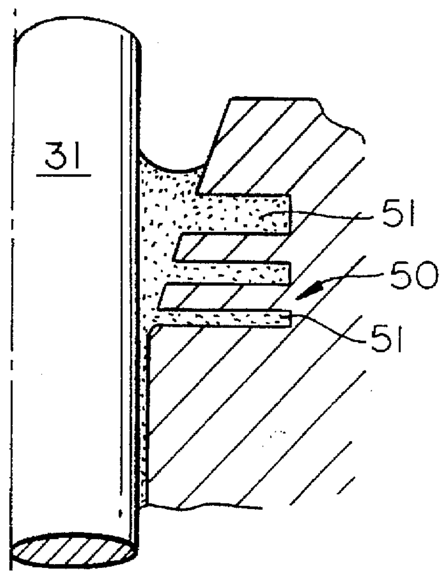


FIG. 4

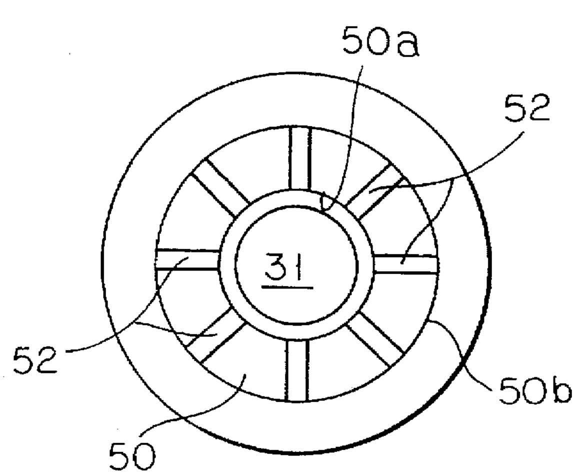


FIG. 5

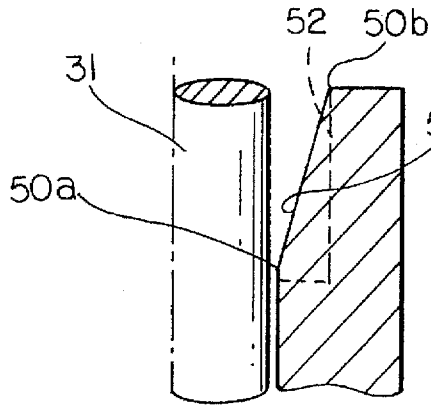


FIG. 6

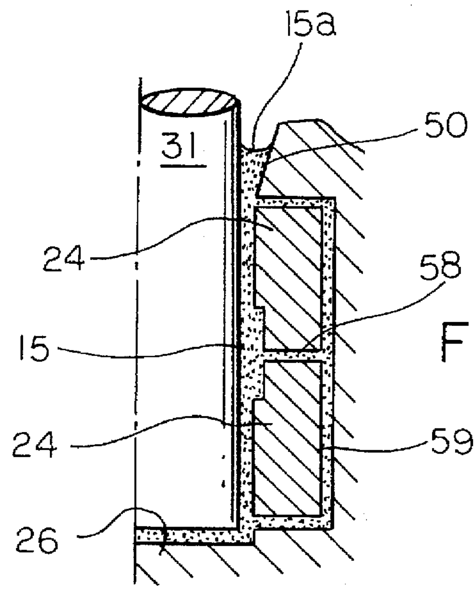


FIG. 7

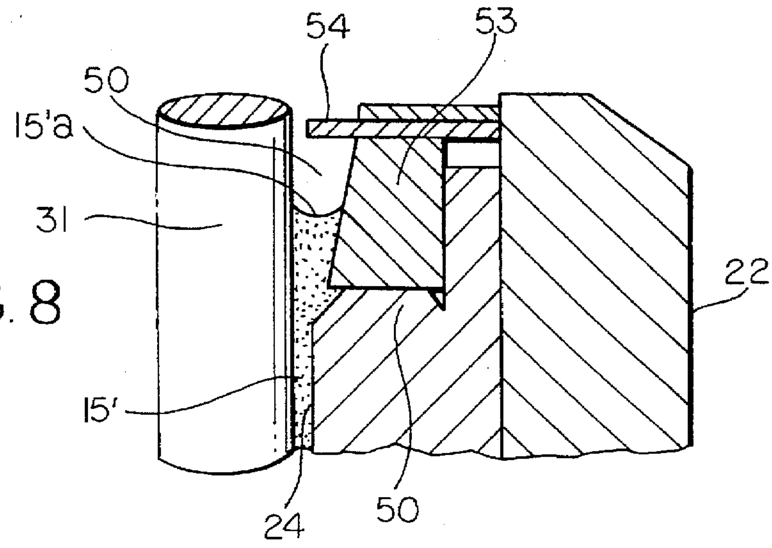


FIG. 8

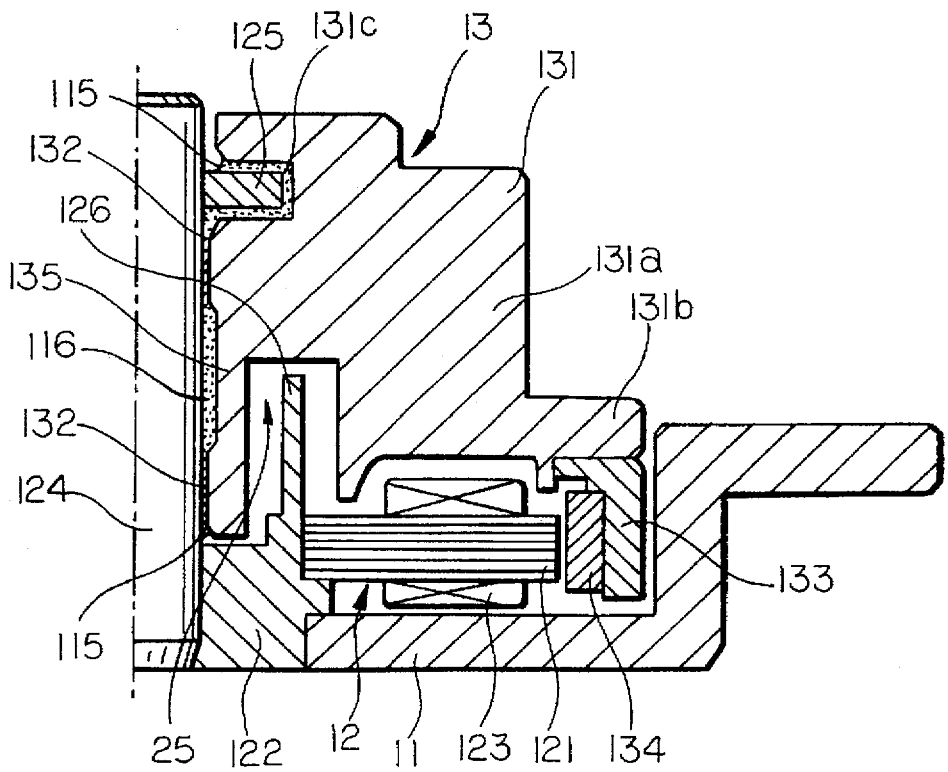

125
131c
13
131
115
132
126
131a
135
131b
116
132
124
133
115
25
122
12
11
123
121
134

FIG. 9

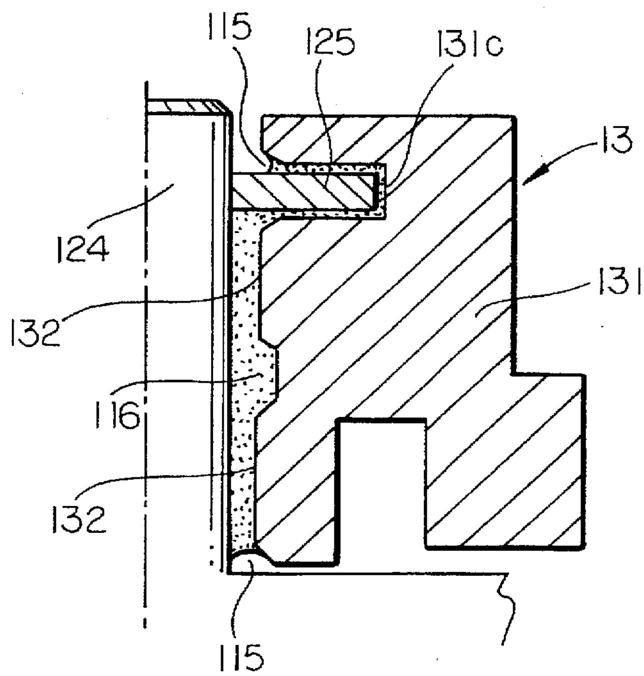
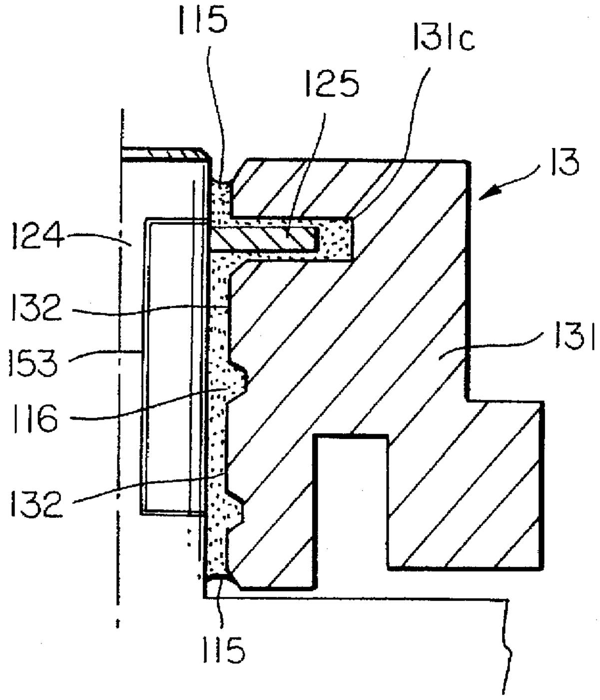
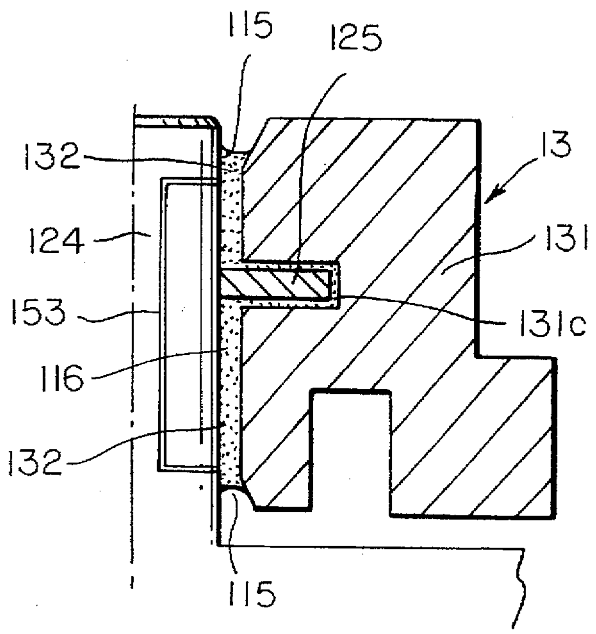

115
125
131c
13
124
131
132
116
132
115

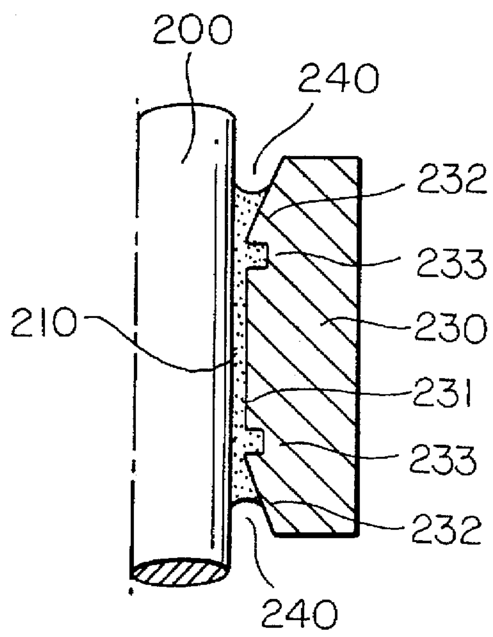
FIG. 10


115
131c
125
13
124
132
153
131
116
132
115

F I G. 11

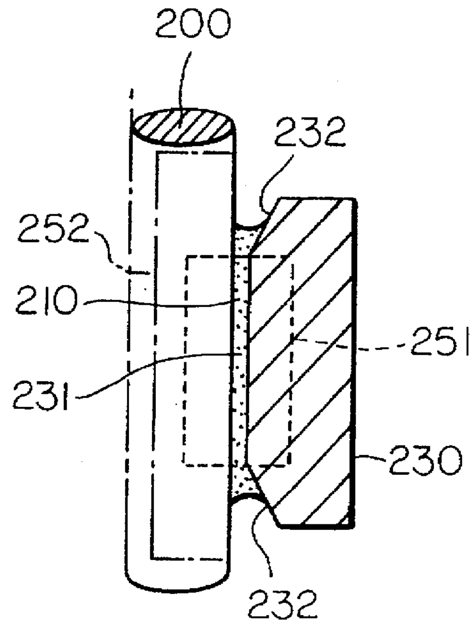

115
125
132
13
124
131
153
131c
116
132
115

F I G. 12



200
240
232
233
210
230
231
233
232
240

FIG.13



200
232
252
210
251
231
230
232

FIG.14

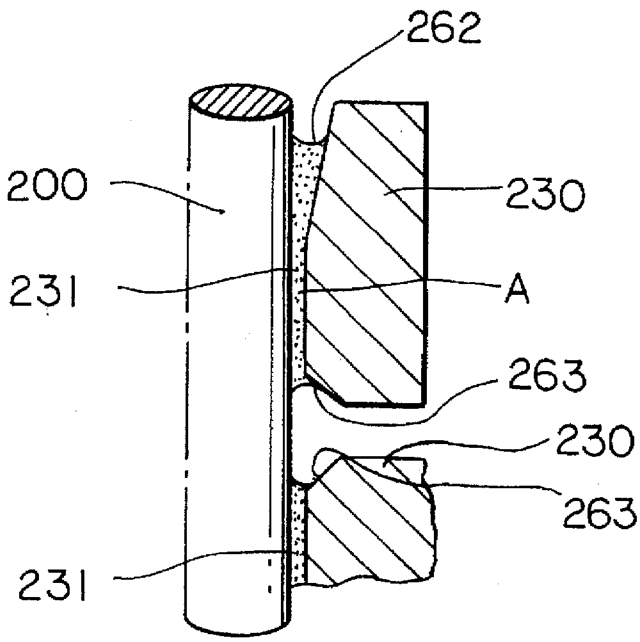

262
200
230
231
A
263
230
263
231

FIG. 15

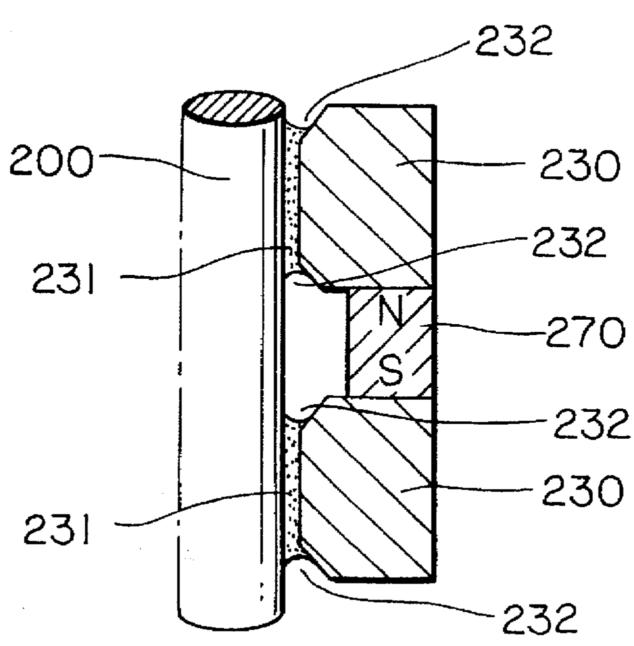

232
200
230
231
232
N
270
S
232
230
231
232

FIG. 16

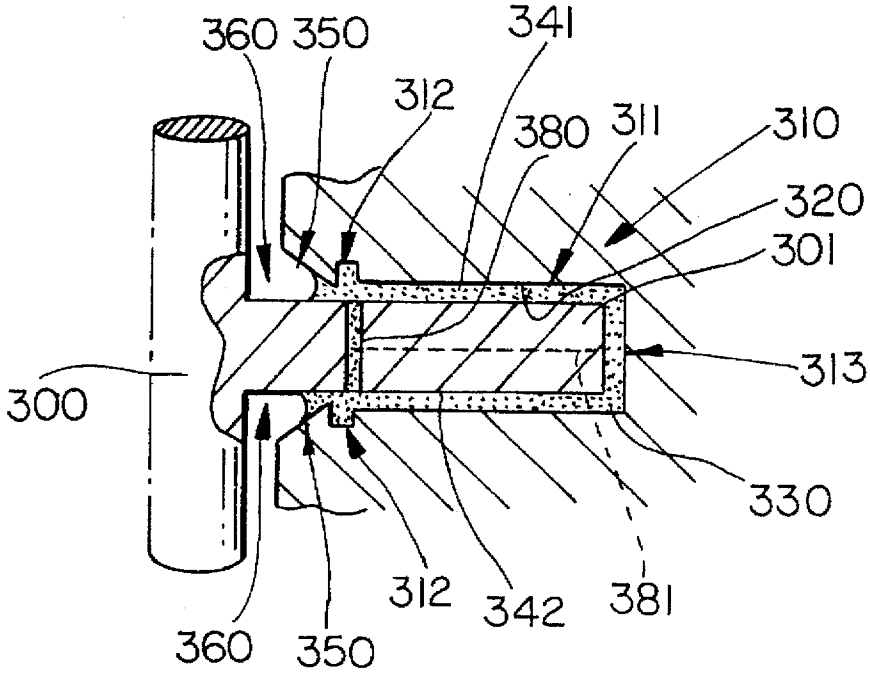

360
350
341
312
380
311
310
320
301
300
313
330
312
342
381
360
350

FIG. 17

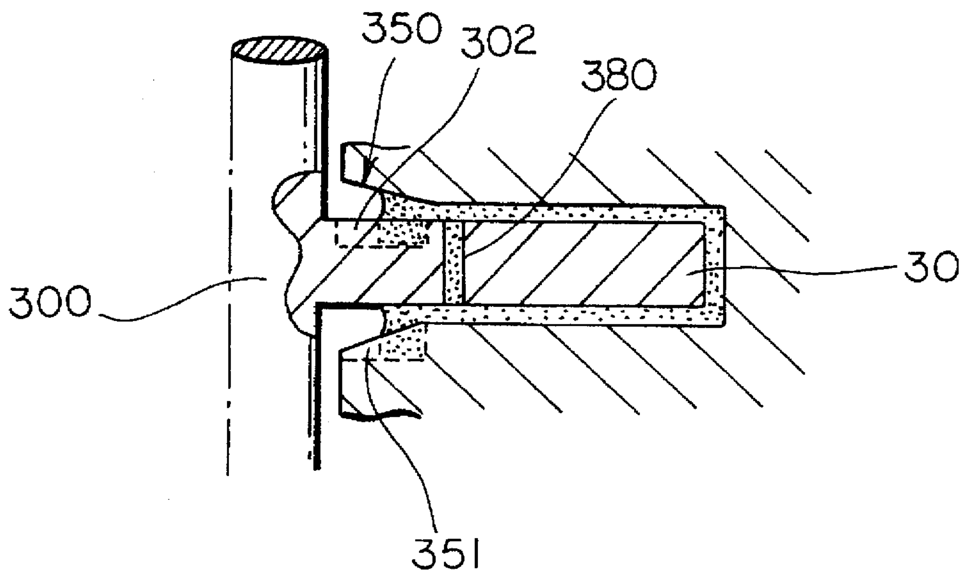

350
302
380
300
301
351

FIG. 18

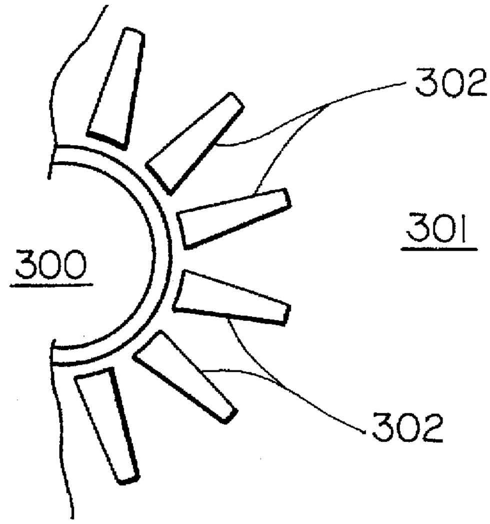


FIG. 19

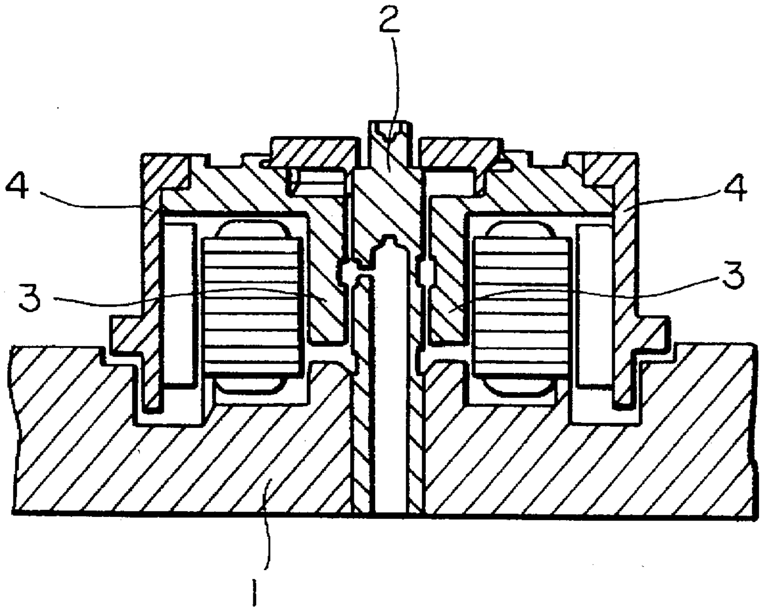


FIG. 20
PRIOR ART

BEARING SEAL SYSTEM

BACKGROUND OF THE INVENTION

a) Field of the Invention

This invention relates to a bearing seal system which prevents lubrication oil from leaking outside the beating section where the oil lubricates a beating to relatively rotatably support a rotating member against a fixed member.

b) Description of the Related Art

In recent years, a variety of bearings using oil as a lubricant such as genetic "slipping bearings" and hydrodynamic pressure bearings have been proposed. As an example of these products using such bearings, a motor for a hard disk drive system (HDD) of the fixed-shaft type is illustrated in FIG. **20**. This motor has a rotatable hub **4** installed on a fixed shaft **2**, which is fixed on a frame **1** via a radial bearing **3**, so that a rotation lubrication oil is supplied to the space between fixed shaft **2** and radial bearing **3**. The oil is retained inside the bearing section of the radial bearing **3** by capillary action.

Various rotation systems, in which a shaft rotates with lubrication oil such as a motor rotation shaft, always require oil leak prevention measures. Particularly, oil leakage is a serious problem for the bearings in HDD motors or laser beam printer (LBP) motors in which cleanliness is critical. The conventional bearing system is one in which oil is adhered only in the space of the bearing section by capillary action or a special sealing mechanism is used to prevent oil from leaking. However, these conventional technologies, for example, hydrodynamic pressure bearings, are unable to obtain a functional lubrication, such as a hydrodynamic pressure, when little lubrication oil is supplied and oil leaks when too much oil is supplied without appropriate oil leak prevention measures. Also in the conventional technology, the bearing seal system does not take into account external forces such as gravity, vibration, impact, centrifugal force, hydrodynamic pressure, atmospheric pressure, temperature and other pressures. This results in a system with poor dependability.

OBJECT AND SUMMARY OF THE INVENTION

It is a primary object of the present invention to provide a bearing seal system which takes into account the following:

(1) a structure wherein a bearing has a space for taking into account the change in oil quantity injected and/or moved and oil is retained stably;

(2) a structure in which oil does not come out when exposed to external forces (gravity, vibration, impact, centrifugal force, hydrodynamic pressure, atmospheric pressure, temperature and other pressures) and is resistant to external forces;

(3) a structure in which oil does not move easily;

(4) a structure in which the oil surface on the exit side, on the outer side from the bearing section is secured and oil leakage is prevented; and

(5) a structure in which oil is not mixed with air easily in order to provide a bearing seal system which:

 (1) retains oil in the beating section at all times to satisfy required bearing properties and

 (2) is oil leak tight.

Before describing examples of this invention, inventive principles that inventors found through their elaborate research are disclosed to assist understanding this invention.

To retain oil in a bearing with the structure of having two opposing exits like that of a general radial bearing, oil is

retained by balancing the capillary suction pressure and its surface position is determined. This status is balanced by two pressures: oil moves to a balancing position when some pressure is added from one side. For example, oil moves from the position A=B to the position wherein (A=B+external pressure), and oil stays where the pressures are balanced.

As such, in the structure having two exits, the oil position in a bearing is determined by pressure balance; therefore,

(1) oil moves whenever external pressure is added; a space is needed to retain oil in order to prevent oil from leaking during moving. It is highly probable that repeated oil movement invites air into the oil;

(2) oil retention pressure generated by capillary action is inversely proportional to the distance between the tapering space section and the shaft because the minimum tapering space section is normally regulated by the tapering space of the bearing section, thus increasing oil retention pressure is limited.

As a result of investigation for resolution of these issues, the inventors concluded that a bearing seal system having a bag structure (in which one of the two exits is closed as if one side of a cylindrical bag is closed) as illustrated in Example 1 provides structural effects as described below; unlike the structure having two exits, the bag in this system acts like a wall with which one atmospheric pressure is generated and is retained on the opposite side:

(1) oil does not move even when external force is added, thus minimizing the oil retention space and decreasing the probability of allowing air into the oil,

(2) oil is retained with as large as 1 atm, thus enhancing the retention property against external pressures.

To realize these goals, a bearing seal system having a bag structure of a first example of this invention is configured as follows:

(1) oil is filled in the tapering space section positioned from the bag section to the exit section;

(2) the space in the tapering space section is kept to 0.8 mm or less and the slope angle of the outer end of the tapering space section is kept to 45° or less. In this way, oil is kept from air and a stable no-leak status is made possible;

(3) the capacity of the tapering space section is made larger than that of a bag section or a radial beating section in order to retain oil in the beating section at all times and to prevent oil from leaking as a result of variations in the quantity of oil injected or in the capacity, i.e., any capacity change due to coming out of a thrust bearing due to rotation or heat generation, and any oil capacity change due to evaporation or air migration;

(4) the space ratio of the inner end and the outer end of the tapering space section is made larger to prevent oil from moving to the bearing section even when air migrates into the oil surface; the pressure difference by the space ratio naturally pushes air out, thus resolving the mixing situation. With the tapering ratio, oil remains in a stable status at any point.

In another example of this invention, the tapering space section where the surface of lubrication oil is injected can be at two places at both ends of a radial bearing section like a generic radial bearing structure or at both ends of a bearing with a radial bearing section and a thrust bearing section.

In another example of this invention, the two oil exits intersect rectangularly against the rotation shaft, creating a quasi-bag structure to take its structural advantage: the

tapering space section is formed at two oil exits and a hole is formed to allow both tapering space sections to communicate with each other in the axial direction, thus equaling the oil pressure and external pressure in order to obtain the same effect as that in the bag structure. This structure entirely supports the pressure generated by centrifugal force with the bag section.

For a better understanding of the present invention, reference is made to the following description and accompanying drawings while the scope of the invention will be pointed out in the appended claims.

BRIEF DESCRIPTION OF THE DRAWINGS

In the drawings:

FIG. **1** illustrates a half cross section of the HDD motor structure of the first example of this invention;

FIG. **2** illustrates a half cross section of the bearing seal system of the first example;

FIG. **3** illustrates an illustration of the major section of the first example;

FIG. **4** illustrates a half cross section of an enlarged modified major section of the first example;

FIG. **5** illustrates an enlarged top view of a modified major section of the first example;

FIG. **6** illustrates a cross section of the bearing seal system illustrated in FIG. **5**;

FIG. **7** illustrates a half cross section of a modified bearing seal system of the first example;

FIG. **8** illustrates an enlarged half cross section of a modified major section of the beating seal system;

FIG. **9** illustrates a half cross section of a HDD motor structure of the second example of this invention;

FIG. **10** illustrates a half cross section of a beating seal system of the second example;

FIG. **11** illustrates a half cross section of a modified example of the second example;

FIG. **12** illustrates a half cross section of another modification of the second example;

FIG. **13** illustrates a half cross section of a beating seal system of the third example of this invention;

FIG. **14** illustrates a half cross section of the modification of the third example;

FIG. **15** illustrates a half cross section of another modification of the third example;

FIG. **16** illustrates a half cross section of one more modification of the third example;

FIG. **17** illustrates a half cross section of a bearing seal system of the fourth example of this invention;

FIG. **18** illustrates a half cross section of a modification of the fourth example;

FIG. **19** illustrates a top view of enlarged major section of FIG. **18**; and

FIG. **20** illustrates a cross section of a conventional bearing system structure.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

The specific examples of a beating seal system of this invention which has been applied to a spindle motor for a HDD hard disk drive will be explained based on the annexed drawings.

A right half of an HDD spindle motor cross section taken from the rotation axis X—X is illustrated in FIG. **1**. The spindle motor is comprised of a stator assembly **20** which acts as a fixed member assembled in a fixed frame **10** on the disk driving system and a rotor assembly **30** which acts as a rotating member assembled in layers against the stator assembly **20** from the upper side is shown in the figure. A stator core **21** constituting the stator assembly **20** is fitted onto the outer peripheral portion of a cylindrical bearing holder **22** installed in the center of the frame **10** and the salient-poles of the stator core **21** are wound with coil **23**.

On the inner peripheral portion of the bearing holder **22**, a radial bearing, comprised of a pair of radial lubrication bearings **24**, arranged at certain intervals in the axial direction, is fixed and the rotation shaft is rotatably supported by the pair of radial lubrication bearings. On the inner peripheral surface of the radial lubrication bearings **24**, a known radial hydrodynamic pressure generating groove is formed, frictionally facing the outer peripheral surface of the rotation shaft **31** via bearing oil **15**, and the hydrodynamic lubrication surface consists of the inner peripheral surface of each of the radial lubrication bearing **24** and the outer peripheral surface of the rotation shaft **31**.

The bottom end of the rotation shaft **31** is supported by the hydrodynamic thrust bearing comprised of a thrust backing plate **25** to cover the opening formed on the bottom end surface of the bearing holder **22** wherein the lubrication surface of the thrust backing plate **22** faces the bottom end of the rotation shaft **31** and a thrust bearing **26** with a thrust hydrodynamic pressure generating groove, as usual, formed on the lubrication surface. Within the thrust backing plate **25**, the radial bearing with a pair of radial lubrication bearings **24** and the rotation shaft **31**, a bag-shaped bag section **40** comprising a cylindrical path is formed; by filling the bearing seal space of the bag section **40** with oil **15**, the rotation shaft **31** is supported freely and rotatably.

A hub **37** consisting of the rotor assembly **30** is bonded to the upper end portion of the rotation shaft in the figure to rotate with the shaft. The hub **37** is comprised of a cylindrical clamper **33** which is equipped with multiple magnetic disks **34** and spacer **33*a*** and **33*b***, and is also equipped with a motor rotor driving magnet **36** on the hub **37** via a back yoke **35**. The driving magnet **36** is circular and is positioned to face closely the outer peripheral end surface of the stator core **21**.

In this embodiment, a bearing section is specifically illustrated in FIG. **2**; a tapering space section **50** is formed outside of the radial bearing **24**; it is positioned on the outside in the axial direction in order to prevent the oil **15** from leaking. Liquid surface **15*a*** of the oil **15**, which acts as a bearing lubrication, fills the bag shaped bearing space which corresponds to the cylindrical bag section **40** (including the thrust bearing section, the radial bearing section, and the space between) and is designed to stay within the tapering space section **50**.

The tapering space section **50** is composed as follows: In the tapering space section **50**, the far inner end in the axial direction on the side of the radial bearing **24** side is the inner end of the tapering space section **50*a***; the section on the side of the radial bearing **24** away from the inner end of the tapering space section **50*a*** is the bag section **40**. The far outer end of the tapering space section **50** is the tapering space section outer end **50*b*** and the outer side of the tapering space section outer end corresponds to the outer side of the tapering space section **50**.

When the angle α of the tapering space section **50** at a predetermined position in the axial direction viewed from the side of the bag section **40** is defined as the slope angle of the tapering space section,

when $\alpha=0°$, the space is parallel to the rotation shaft surface, when the angle a which increases its degree to the outer side is defined as plus, and when the angle α which increases its degree to the bag section **40** is defined as minus, the space at the tapering space section **50** is set as follows:

the narrowest is the inner end of the tapering space section **50***a*, the widest is the outer end of the tapering space section **50***b*, and the slope angle α of the tapering space section from the tapering space section inner end **50***a* to the tapering space section outer end **50***b* is 0° or more. When the slope angle α of the tapering space section is 0°, the area parallel to the rotation shaft **31** may exist in pan of the area of the tapering space section **50**.

When a space with the slope angle minus α exists between the tapering space section and the bearing section, the tapering space section inner end **50***a* corresponds to the far outer end in the axial direction of the space where the slope angle α of the tapering space section becomes minus for the first time. The space at the outer end of the tapering space section needs to be 0.8 mm or less; when the space has a dimension is over 0.8 mm and the slope angle α of the tapering space section at the point where the space dimension is 0.8 mm is 45° or more, the widest space which satisfies the condition of 0.8 mm or less and 45° or less is defined as the tapering space section outer end **50***b*.

In this example, the capacity of the tapering space section **50** (the capacity created with the tapering space section inner end **50***a*, the tapering space section outer end **50***b*, and the rotation shaft **31**) is set as 5% or more of the entire capacity of the bag section **40** inside the border of the inner end of the tapering space section **50** and 100% or more of the capacity of a pair of radial bearings **24** (the capacity created between the radial beating **24** and the rotation shaft **31**); and the space dimension at the outer end **50***b* of the tapering space section **50** is set to be twice or more of that at the inner end **50***a* of the tapering space section **50**.

The function of the bearing seal system in this invention comprising the structures described above will now be explained, referring to FIG. **2**. Because the bearing seal system adopts a bag structure as a basic structure comprising the bag section **40** at which the cylindrical part is covered with the thrust backing plate **25**, the bottom section (on the side of the thrust bearing) in FIG. **2** can be regarded as a wall where 1 atm pressure is generated and retained; therefore, the oil **15** does not move even when an external force is added from the top to the thrust bearing side. Therefore, there is no need to take into account oil movement by external force, thus allowing the oil retention space of the tapering space section **50** to be smaller. Since oil movement by external force does not occur, air migration through oil movement does not occur; and as a result, the bearing seal system retains the oil with an amount of pressure as large as 1 atm so as to provide a high external force resistance retention.

The bearing seal system prevents air migration into the oil **15** and provides the stability and the leak tightness because the space from the bag section **40** to the tapering space section **50** is filled with the oil **15**; the space between the tapering space section **50** and the rotation shaft **31** is set to be 0.8 mm or less; the slope angle α of the tapering space section at the tapering space section outer end **50***b* is set to be 45° or less; the space dimension at the tapering space section outer end **50***b* is set to be twice or more of that at the inner end **50***a*. Due to the large space ratio between the tapering space section inner end **50***a* and the outer end **50***b* with the slope angle α of the tapering space section, even if the air migrates into the oil surface **15***a* of the tapering space section **50**, the air does not reach the beating, also due to the pressure difference associated with the space ratio, the air naturally moves out, eliminating the problem associated with "air migration."

As described, because the capacity of the tapering space section **50** is set larger than that of the bag section **40** or the radial bearings **24**, the oil **15** is always retained in the bearing and prevented from leaking regardless of:

the variation in the quantity of injected oil or the manufacturing variation in the internal capacity of the bag section **40**;

oil coming out of the thrust bearing surface during the rotation of the rotation shaft **31** or the change in the capacity of the bag section **40** due to heat generation during the rotation, or

the change in quantity of the oil **15** due to evaporation or air migration. A groove **27** can be formed to store the oil inside the tapering space section **50** in order to resolve the variation of the quantity of the injected oil and the manufacturing variation in the internal capacity of the bag section **40**.

In this example, when the capacity of the tapering space section **50** is A, the quantity of oil to be injected inside the bag section **40** constituting the bearing space is set to stay within the level between 0.1 A and 0.9 A measured from the tapering space section inner end **50***a* in a stable stationary state.

Basically, when the oil surface **15***a* is inside the tapering space section **50**, the oil remains stable; however, even if the quantity of oil (the level of the oil surface **15***a*) varies with lapse of time or by change in environment, the oil shortage or leakage can be complemented by filling the oil to reach the point (between 0.1 A and 0.9 A) inside the tapering space section **50** so as to maintain the excellent performance under the normal environment.

As illustrated in FIG. **3**, it is recommended that the contact angle θ_1 of the oil **15** at both surfaces of the rotation shaft **31** and the tapering space section **50** are set to be 15° or more. When the oil surface **15***a* is positioned inside the tapering space section **50**, the oil **15** contacts the rotation shaft **31** and the tapering space section **50** with certain contact angle θ_1, and the contact angle θ_1 is set to be 15° or more.

As a result of various experiments, the inventors found that the wet dilation (climbing-up phenomenon of liquid) needs to be prevented in order to prevent the oil **15** from leaking and to obstruct the path, which will be explained next.

In order to prevent the wet dilation caused by any small changes in environment and conditions, the condition of (γS<L+SL) always needs to be satisfied.

Regarding the contact angle of the solid surface and the liquid, the condition where the solid surface and the liquid are maintained in equilibrium is:

$$\gamma S-\gamma SL=\gamma L\cos\theta 0 \qquad \text{(Young's equation)}$$

wherein

γSL: the tension of the solid-liquid interface,

γS: the surface tension of the solid,

γL: the surface tension of the liquid, and

θ1: the contact angle of the solid and the liquid

The equilibrium is determined by the balance of the three surface tensions.

The value of ($\gamma S-\gamma SL$) is the key in that the energy

1) decreases ($\gamma S>\gamma SL$),

2) increases ($\gamma S<\gamma SL$), or

3) remains the same ($\gamma S=\gamma SL$) by replacing the solid surface with the solid-liquid interface.

When the energy decreases (1), the surface is wet, that is, stable at the the solid-liquid interface; when the energy increases (2) or remains the same (3), the surface is not wet, that is, stable at the solid surface.

The difference between γS and γSL is ($\gamma L \cos \theta 1$) where the contact angle $\theta 1$ of the solid and the liquid balances the equation. That is, the bigger the value of ($\gamma S-\gamma SL$) is, the smaller $\theta 1$ is; at ($\gamma S>L+\gamma SL$), the balance is destroyed, even if the condition of the contact angle of the solid and the liquid, $\theta 1=0°$, and the liquid is spread all over the solid surface. This is the same phenomenon that the oil as liquid climbs up the shaft surface as solid, the same as oil dropped on water spreads over the water.

To solve the problem that oil climbs up the shaft surface depends on whether the condition is ($\gamma S>\gamma L+\gamma SL$) or not. This equation shows that the energy decreases and the surface becomes more stable when the interface (γSL) of the solid and the liquid and the liquid surface (γL) are newly created on the solid surface (for example, of a shaft) than when there is only the solid surface (γS). In this case, so-called solid surface is eliminated and the the solid-liquid interface and the liquid surface are newly created; the climbing-up wet dilation can not be stopped.

Even if the external forces (e.g. gravity, vibration, impact, centrifugal force, magnetic force, or other pressures) are added to the condition of above issue, these external forces work in the direction which changes the curvature of the liquid surface in a form of consequent pressure and does not change the equilibrium point. These forces try to move the position of the equilibrium point with the surface tension of the liquid, but have no effects under the climbing-up phenomenon.

As long as the condition of ($\gamma S>\gamma L+\gamma SL$) is maintained, the climbing-up (wet dilation) phenomenon of liquid can not be prevented: the condition needs to be changed to ($\gamma S<\gamma L+\gamma SL$) in order to prevent the climbing-up phenomenon. Specifically, the surface tension of the solid surface needs to be decreased. Under the condition ($\gamma S>\gamma L+\gamma SL$), the external forces have effect via the surface tension of the liquid. Generally, a metal surface obtains a very large surface tension. Usually a formed layer of pellicle decreases the surface tension; however, since the tension is still large, the condition ($\gamma S>\gamma L+\gamma SL$) occurs and causes the climbing-up phenomenon of liquid.

The ways to prevent the climbing-up phenomenon are:

1) setting the condition to be ($\gamma S<\gamma L+\gamma SL$),

2) enlarging he actual contact angle of the solid surface and the liquid, and

3) using external forces efficiently.

To implement 1), 2), and 3):

1) maintain the condition of ($\gamma S<\gamma L+\gamma SL$) by protecting the surface with low surface tension material such as oil repellant agent or the like so as not to expose the metal surface directly,

2) minimize surface roughness and enlarge the actual contact angle of the solid surface and the liquid; the solid surface needs to be free of gaps, grooves, scratches, and irregularities; the wider the surface area is, the smaller the actual contact angle is, and

3) have external forces work in the direction to pull back the climbing-up phenomenon.

The larger the contact angle $\theta 1$ of the solid and the liquid is, the less the oil tends to leak (the stronger the retention). In order to reach this condition, the contact angles $\theta 1$ of the oil **15** with both surfaces of the rotation shaft **31** of the tapering space section **50** and the tapering space section itself needs to be set as 15° or more. Specifically, these conditions are satisfied by placing a relatively low surface tension material such as plastic on the surface in contact with the liquid. If the inner wall surface of the tapering space section **50** is constituted with a low surface tension plastic material which is reasonably immune to wet dilation and the oil surface movement by capillary action, chemically stable, and highly processable, a practical product will be provided. The plastic material can be put on the inner wall surface of the tapering space section **50** by the way of coating or painting.

Furthermore, it is recommended that the difference between two contact angles $\theta 1$ of the oil **15** with both the surface of the rotation shaft **31** of the tapering space section **50** and the tapering space section **50** itself is set to be 15° or less because the smaller the angle difference between the two angles $\theta 1$ is, the less the oil tends to leak.

In this example, the surface roughness Ra of the inner wall of the tapering space section **50** is set to be 0.25 μm or less to minimize the surface roughness of the solid. If the inner wall surface of the tapering space section **50** is rough, the roughness causes the same condition as capillary action. The capillary action occurs when the ratio of the solid surface which the oil surface contacts is large as compared to the oil volume; the same phenomenon occurs when the irregular roughness or grooves exist on the surface. When the contact angle $\theta 1$ of the solid and the liquid is less than 90° ($\theta 1<90°$), the actual contact angle becomes smaller and the oil becomes susceptible for leakage if irregular roughness or grooves exist on the surface. It is possible to increase the actual contact angle of the solid and the liquid to prevent the oil from leaking by reducing the surface roughness Ra of the inner wall of the tapering space section **50.** The configuration that the surface roughness Ra is set to be 0.25 μm or less can be applied to the side of the rotation shaft **31** if needed.

Actually, the contact angle of the oil varies depending on the solid surface condition; and once being wet, the surface becomes easier to be wet again than the surface which has not been wet and the contact angle of the oil becomes smaller. For example, when a water drop moves on the slant and dirty surface of a glass, the contact angle of the water drop on the front side of the glass is large; the contact angle becomes smaller after the water drop moves. The surface once wet is easier to be wet again and the water drops through the same path on the surface due to the difference in the microscopical shape of the solid surface or the difference of surface tensions. That is, if the surface is irregularly rough even when the contact angle is same, the curvature is radically changed with the slope of the solid surface and it is found that the contact angle maintains the balance in the different condition macroscopically. When the non-uniform surface tensions exist, the oil in concave space or surrounded by dirts remains on the surface even after the entire oil withdraws; when the oil climbs up again, the contact angle is unable to grow macroscopically and communicates with the oil left before. As a result, the contact angle becomes smaller and the oil tends to move the same path on the surface as before.

With such a solid surface, when the surface is once wet for any reason or contacted with the oil during injection, its contact angle and retention are smaller and easily move the

oil. The irregular surface roughness and dirt (the non-uniform surface tensions) at the bearing outlet, that is, the outer side of the tapering space section, needs to be reduced as much as possible.

It is said that the macroscopic (apparent) contact angle of the solid and the liquid when the solid surface has an irregular roughness and the difference of the surface tensions becomes as follows:

With an irregular surface roughness, when the surface area ratio of the actual surface to the smooth (flat) surface is r, the macroscopic (apparent) contact angle θW (Wenzel contact angle) is

$\cos \theta W = r \cos \theta 1$ ($\theta 1$: Micro (actual) contact angle).

If r is 2 or more for the irregular surface roughness, the macro contact angle becomes $\cos \theta w = 0°$, even when the micro contact angle is $\theta 1 = 60°$ ($\cos \theta = 0.5$), which is considered as large and the climbing-up phenomenon of liquid (wet dilation) cannot be stopped. It is recommended that the surface roughness of the surface from the tapering space section to the outer side is reduced and is polished like a mirror in order to prevent an oil leakage.

On the other hand, the surface with the difference of the surface tensions can be regarded as a composite surface of different surface tensions; therefore, the macro (apparent) contact angle θC (Cassy contact angle) is:

$$\cos \theta C = A1 \cos \theta 1 + A2 \cos \theta 2.$$

A1 and A2: the ratio of the surfaces occupied by the different surface tensions and $\theta 1$ and $\theta 2$: the micro (actual) contact angle of different surface tensions.

It is also recommended to reduce the roughness of the surface from the tapering space section to the outer side and polish it like a mirror in order to prevent an oil leakage.

When any point within ½ length from the outer end **50***b* in the axial direction of the tapering space section **50** is the home position, it is effective to form a tapering space section **50**, using the material or the surface processing, with the contact angle $\theta 1$ at least 15° larger at the inner wall surface covering the home position to the outer side than that covering the interior side of the home position. To enlarge the contact angle of the solid and the liquid is one of the methods to prevent the oil leakage. When the contact angle cannot be enlarged for any reason, it is effective to enlarge only the most important contact angle for retention.

Furthermore, it is effective that the contact angle at the outer end **50***b*, that is, at the larger radius side of the tapering space section **50**, is set larger than that at the inner end **50***a*, that is, at the smaller radius side. Because the oil inside the bearing is affected by the centrifugal force while the rotating member rotates, the oil pressure of the surface on the larger radius side in a radius direction of the tapering space section is larger than that on the smaller radius side. When the contact angles of the surfaces on both the inner and the outer ends are same, the contact point of the solid and the oil surface on the larger radius side is positioned outside of the contact point of the solid and the oil surface on the smaller radius side and the surface becomes susceptible for oil leakage and air migration. When the contact angle on the larger radius side is enlarged, the contact point of the solid and the oil surface on the larger radius side becomes closer to the contact point of that on the smaller path side, and the surface excels in oil leakage and air migration.

In the example of this invention, the outer end (the upper end in FIG. **2**) of the hydrodynamic pressure generating groove of the radial bearing **24** can be extended to

the oil reservoir groove **27** when created as shown in FIG. **2** or

the inner end **50***a* of the tapering space section **50** when the oil reservoir groove was not created. If the hydrodynamic pressure generating groove is extended to the inner end **50***a* of the tapering space section **50**, the slope angle α of the radial bearing **24** is always maintained as $\alpha > 0°$, resulting in prevention of air coming in during oil injection and the oil being retained with force in the direction to push out migrated air if any to always maintain conditions which retain oil easily.

The tapering space section **50** is formed to create an opening with the angle of 45° or less viewing from the inner axial direction side to the opening side; the slope angle of the outer tapering space section **50** is at 45° or less. With this structure, the condition between the solid surface of the tapering space section **50** and the oil **15** is secured and the oil leak cage is prevented even if the surface of the oil **15** rises over the expected point at the tapering space section **50**.

When the slope angle α of the tapering space section **50** is fixed and formed on the inner wall surface being perfectly flat in cross section, it is the easiest shape to be processed and the oil becomes stable with force to absorb the oil inside and push out the air because the slope angle a of the tapering space section is more than 0° at any point.

It is recommended that the average slope angle of the tapering space section **50** is set to be 10° or more. An average slope angle of more than 10° is needed to prevent the oil transfer caused by external force or the space change by a relative move.

It is possible that more than two thirds of the space of the tapering space section **50** in the axial direction is set parallel at a distance of 0.4 mm or less (the slope angle $\alpha = 0°$). With this shape, wider space for the tapering space section **50** is available and the difference and variation of the oil capacity **15** can be resolved, the distance between the parallel tapering space can be reduced; and a leak tight condition is secured.

The example in FIG. **4** illustrates that a space **51** is created in the radial direction inside the tapering space section **50**. By creating the space **51** in the radial direction, the capacity for the oil retention can be increased; by maintaining the smaller width of the space **51** in the axial direction than that between the outer end **50***b* of the tapering space section **50** and the rotation shaft **31**, the oil in the space **51** is firmly retained. Because the space **51** formed in the radial direction does not take much space in the axial direction, the total width in the axial direction can be reduced and the impact resistance retention becomes effective.

If the condition of the tapering space section **50** described before is satisfied, the space **51** can be created in the axial direction like the groove-like space created in the radial direction or as a hole, as illustrated in FIGS. **5** and **6**.

As illustrated in FIG. **7**, a circulation hole **53** can be formed to connect two radial bearings **24** via the outer side of each radial bearing **24**. Because the hydrodynamic pressure generated in the radial hydrodynamic bearing is large, a pressure is generated to cause an oil leak when the hydrodynamic pressure is unbalanced. However, the pressure difference caused by the hydrodynamic pressure can be resolved and the oil leakage is prevented by connecting two radial bearings **24** via the outer side of each radial bearing **24** with the circulation hole **53**. The retention pressure by the bag structure is utilized effectively at the tapering space section **50**.

A groove extending in the shaft direction can be formed from the tapering space section **50** to the outside. This structure creates the condition where the exchange of the oil **15** and the air is smooth when the air migrates into the oil

15 for any reason and the air is pushed out with much more certainty. The groove pushes the oil to the narrower space and the air to the wider space, and separates the air and oil. If the groove is extended in the outer axial direction, the separated oil or air is able to move along the groove for smooth exchange.

The ratio of air to oil in the bearing space should preferably be 2% or less. When oil contains air, the volume of air is inversely proportional to pressure and proportional to the absolute temperature because it is a gas. Therefore, if the ratio of air in relation to oil is not maintained at less than a certain level, oil may leak or be short due to a change in pressure or temperature. When the ratio can be kept at 2% or less using the vacuum-injection method or the like, the oil capacity containing migrated air increases by 2% or less at 0.5 atmospheric pressure, and by 0.4% or less when the temperature is raised by 60° C.; those changes do not cause leakage due to the capacity ratio of the bag section and the tapering space section **50**.

In the example illustrated in FIG. **8**, a magnetic fluid **15'** is used as oil and a (magnetized) magnet **53** is located in the radius direction in the tapering space section **50** to form a magnetic circuitry between the tapering space section and the rotation shaft **31**. A seal plate **54** is located outside the magnet **53**. The magnetic circuitry is strong because the space between inside the tapering space section **50** and the rotation shaft **31** is narrow and is weak at the outer end because the space between the outer end of the tapering space section **50** and the rotation shaft **31** is wide, and, at the same time, a magnetic shield of predetermined magnetic flux density gradient is generated in almost the same direction within more than a half area of the tapering space section **50**.

By establishing these magnetic conditions and constructing the tapering space section **50** as described above, the magnetic fluid **15'** obtains the inward force not only from the tapering space section **50** but also from the magnetic force, resulting in creation of the more leak tight status. By maintaining the magnetic flux density gradient constant, a predetermined level or more of magnetic force is added even though the surface position **15'***a* of the magnetic fluid **15'** changes to some extent.

The second example in which the bearing structure is configured differently from the first example is illustrated in FIG. **9**.

In the center of a core holder **122** supporting a stator assembly **12**, a fixed shaft **124** which acts as the core during motor rotation, is perpendicularly installed toward upward in the figure; a hub **131** constituting a rotor assembly **13** is rotatably supported against the outer peripheral of the fixed shaft **124** via a pair of radial lubrication bearing sections **132**. The hub **131** comprises a cylindrical body **131***a* for a magnetic disk installation and a mount **131***b* located at the bottom edge of the body end of **131***a* as illustrated, and a motor driving magnet **134**, which is a motor rotor magnet, is installed on the mount **131***b* via a back yoke **133**.

The middle section **135** of the radial lubrication bearing sections **132** can be constituted as a single component or a cylindrical spacer can be inserted within it; both radial lubrication beating sections **132** are formed at a predetermined distance in the shaft direction. Each inner peripheral surface of radial lubrication bearing **132** and outer peripheral surface of the fixed shaft **124** mutually constitute lubrication surfaces, and a predetermined amount of bearing oil is filled in a cylindrical beating section **116** including both the lubrication surfaces.

In the upper end section of the fixed shaft **124** illustrated, a thrust plate **125** which constitutes a thrust beating is installed and a circular recess section **131***c* is formed on the hub **131** side to hold the thrust plate **125**. The circular recessed section **131***c* is continually filled with the bearing oil from the bearing section **116**; the thrust plate **125** can rotatably support the rotating body including the hub **131** in the circular recessed section **131***c*.

A circular projection **126** formed on the core holder **122** is extended in the axial direction (upward in the illustration) by a predetermined length and the]-shaped (cross section) narrow path **25** is formed by each outer peripheral wall surface of the radial lubrication bearing **132** and the middle section **135** and the inner peripheral wall surface of the hub **131**, providing a leak-tight structure.

In this example, as illustrated in FIG. **10**, the tapering space sections **115**, described in FIG. **2** are located in two places at both sides of the cylindrical bearing **116**. The beating oil is continually filled from inside the bearing **116** through the two tapering space sections **115**, so that the surface position of the outer end of oil stays inside the tapering space section **115**. See previous examples for details so that the conditions in each tapering space section **115** remains the same as described above.

In the example illustrated in FIG. **10**, the capacity of the tapering space sections **115** is set to be 10% or more of that between the inner ends of both tapering space sections and 100% or more of the radial bearing sections. The ratio of the dimension of the outer end of the tapering space section **115** to that of inner end of the tapering space section **115** is set to be more than 2.

In this example also, both tapering space sections **115** were continually filled with oil and are configured as described above, so the structure does not easily allow air to migrate into oil and provides a stable leak-tight status. Against variation in the volume of injected oil or internal capacity, the capacity change due to a thrust bearing's coming out during rotation or the like, heat generation, evaporation or air migration, oil is always retained in the bearing section and is prevented from leaking.

Regarding the direction of the opening of the tapering space section **115**, the tapering space section **115** at upper side is opened toward the shaft: rotation generates a centrifugal force to the oil which is always larger at the larger radius side; therefore, this configuration prevents the pressure by centrifugal force to work in the direction that allows oil to leak, providing a stable status. The tapering space section **115** at bottom side in FIG. **10** is opened to the direction parallel to the rotation shaft.

The thrust bearing system in this invention is configured to obtain the hydrodynamic pressure in the direction that cancels centrifugal force. Both centrifugal force and hydrodynamic pressure are generated when rotation begins and can balance in the configuration.

In the example illustrated in FIG. **11** or **12**, an oil circulation hole **153** which connects both ends of the bearing **116** is created inside the fixed shaft **124**. The oil circulation hole **153** communicates the inner end side of the upper tapering space section **115**, the outer end of the thrust bearing section which is comprised of a thrust plate **125** and the circular recess section **131***c*, the inner end side of the bottom tapering space section and outside the radial bearing section **132**. In this way, pressure generation is cancelled via a circulation hole **153** when the hydrodynamic force is unbalanced inside the bearing.

The third example of this invention is described referring to FIG. **13**. In the example illustrated in FIG. **13**, oil **210** is filled in the space between the external surface of a fixed shaft **200** and the inner peripheral surface of a radial bearing

230, and the external surfaces at both ends in the axial direction of oil 210 are exposed to air.

The space between the outer peripheral surface of the fixed shaft 200 and the inner peripheral surface of radial bearing 230 is comprised of a radial bearing section 231, the tapering space sections 232, and two external surfaces 240 which are formed in outer shaft direction of a tapering space section 232.

As described above, both ends of the bearing sections 231 have two tapering space sections 232, oil 210 is continually filled from inside the bearing section 231 to both tapering space sections 232 and oil surface 210 is positioned within a tapering space section 232. The tapering space section 232 is constituted under the same conditions as described in previous examples.

Also in this example, the capacity of both tapering space sections 232 is set to be 200% or more of that of the bearing section 231 which exists between both inner ends of the tapering space sections. The distance between the outer end of the tapering space section 232 and the shaft is set to be twice or more of that between inner end of the tapering space section and the shaft.

As illustrated in dashes in FIG. 14, when an oil circulation hole 251 is created to connect both inside of the inner end of the tapering space sections 232 or, as usual, the radial bearings 230 are installed in two places, an exhaust means consisting of a hole 252 to push out the internal air between the radial bearings 230 can be created in the fixed shaft 200 or a space can be created in the radial bearings 230 to connect the radial bearings 230. When air exists in the space between the bearings, air can be expanded or pressurized due to a change in atmospheric pressure or temperature. In this case, the air pressure can push out the intermediary oil in the bearing section. When an air exhaust means consisting of a hole or space communicates with the bearings, the expanded air escapes via the exhaust means, thus canceling the pressure difference, and, as a result, the pressure that push out oil is equalized.

In the example illustrated in FIG. 15, two radial bearings 230 are positioned on both upper and lower side, the slope angle of the outer tapering space section 262 at the inner side is set to be larger than that at the outer side, and at the same time, the oil contact angle remains 45° or less. The smaller the contact angle that generates the maximum retention pressure, the larger the generated pressure is: about 70% of the (maximum retention) pressure is generated at a 45° contact angle and the contact angle should be kept at this level.

Also in the example illustrated in FIG. 15, the average slope angle of the tapering space section 262 and 263 is constructed to be 10° or more. A slope angle of 10° or more on average is required in order to prevent oil from moving when the system is exposed to a change in tapering space due to external forces or relative movement.

As in the example illustrated in FIG. 15, when air exists in the space between the two bearings 230, it is recommended that the average slope angle of the tapering space section 263 positioned at the inner side of the bearing section 231 is set to be twice or more of that at the outer side of the bearing section 231. In this way, the capacity of the tapering space section 262 at the outer side remains larger than that at the inner side, increasing projected actual oil retention capacity. Oil leaks easily out of the inner tapering space section 263 at the inner side; however, this cannot be a problem because the system returns the oil inside the bearing.

In the example illustrated in FIG. 16, a magnetic fluid is used as oil and a magnet 270 is positioned between two

bearings 230 made of magnetic material to form a magnetic circuitry with the fixed shaft 200 also made of the magnetic material. The magnetic circuitry, provided that a bearing 230 and a fixed shaft 200 are comprised of magnetic materials, is strong at the inner end of the tapering space section 232 and weak at the outer end of the tapering space section 232; the system is designed to form a predetermined magnetic flux density gradient in one direction in at least half the area or more of the tapering space section 232.

This magnetic condition makes it difficult for a magnetic fluid acting as oil to leak due to the magnetic force working inward. By maintaining a magnetic flux density gradient constant, a predetermined level or more of the magnetic force is added even if the position of magnetic fluid is changed slightly.

The fourth example is described in FIG. 17. The example in FIG. 17 illustrates the structural advantage of bag structure using a quasi-bag structure for the condition of two tapering space sections. In this example, spaces used as oil reservoirs communicate with a central hole to equal pressures, and at the same time, the two tapering space sections are positioned in almost the same position in the radius direction to apply almost the same external pressure, obtaining the same effect as the bag structure. The bag section provides the advantage to fully support the pressure from centrifugal force.

In FIG. 17, a hub is installed rotatably on a fixed shaft 300 via a radial bearing, which is not illustrated, and a thrust bearing 310. A thrust bearing 310 is comprised of a thrust disk 301, which is united with the fixed shaft 300 and is inserted into a cylindrical groove 320 extended in the radial direction and grooved on the hub side, and oil 330 is continually filled in the space between the cylindrical groove 320 and the thrust disk 301. In a thrust bearing 310, at least one of the four facing surfaces, created by a cylindrical groove section 320 and the thrust disk 301, has a normally grooved hydrodynamic pressure generating groove.

In the thrust bearing 310, on the smaller radius side of two spaces 341 and 342 formed by four facing surfaces in the shaft direction, created by the groove 320 and the thrust disk 301, two tapering space sections 350 which open to outer passage along the fixed shaft 300 are created: each of the tapering spaces 341 and 342 of the thrust bearing are comprised of the thrust bearing section 311, the tapering space section 350 at the smaller radius side of the bearing section 311, oil reservoir formed between the bearing sections 311 and the tapering space section 350, the outer space 360 formed on the smaller radius side of the tapering space section 350, and the inner space of space 313 formed on the larger radius side of the tapering space section 350 which communicates with the two tapering spaces 341 and 342.

Oil 330 is continuously filled from one tapering space section 350 to the other tapering space section 350, and at the same time the oil reservoirs 312 are communicated with each other in the shaft direction via a center hole 380. The tapering space section 350 is formed to fulfill the conditions described in other examples.

In this example, the total clearance in the shaft direction at the bearing sections 311 is set to be 200 μm or less and the total capacity of the tapering space sections 350 is set to be 100% or more of that of the thrust bearing 310; the total capacity of the tapering space sections 350 is set to be 30% or more of that of the bearing section 311, the oil reservoir 312, the outer space 313, and the center hole 380 and the dimension between the outer end of the tapering space section and shaft is set to be triple or more of that between the inner end of the tapering space section and the shaft.

Due to the total clearance in the shaft direction at the bearing section being is set to be 200 μm or less, the movement in the thrust direction is suppressed and the change in the tapering space which affects oil is suppressed to obtain an excellent oil retention.

As illustrated in dashes in FIG. **17**, an oil circulation hole **381** can be T-figured so the outer space **313** and the oil reservoirs **312** can be connected. By connecting both sides of the thrust bearing section **311** with the circulation hole, the difference in pressure is suppressed to prevent oil leakage.

In the example illustrated in FIGS. **18** and **19**, a tapering space **302**, which satisfies the condition of the tapering space section, is added in the axial direction to the thrust disk **301** which faces the tapering space section **350** in the upper side of FIG. **18**. By creating the tapering space **302** in the axial direction, the capacity to retain oil can be increased; especially by making the space **302** smaller than the outer end width of the tapering space section **350**, oil in the tapering space **302** is retained for certain. The tapering space **302** in the axial direction does not take any space in the thrust beating direction; therefore, the dimension in the thrust bearing direction can be reduced and a better shock resistance retention is retained. The tapering space **302** can be created on only one side of the tapering space sections **350**, and as illustrated together in FIG. **18**, the tapering space **351** in the axial direction can be formed in the tapering space section **350**.

As described above, the bearing seal system of this invention retains oil in the bearing section constantly and sufficiently, and at the same time, retains oil in a stable manner, demonstrating its excellence in the prevention of oil leak cage and external force resistance so as to improve the dependability of the system.

Due to the larger capacity of the tapering space section as compared to that of the bag section, the radial bearing section, and the thrust bearing section, the oil is always retained in the bearing section and prevented from leaking regardless of variations in injected oil or the internal capacity, capacity change caused by the thrust bearing coming out to the surface during rotation or the heat generation, or change in oil capacity due to evaporation or migrated air.

The large space ratio of the inner end and the outer end of the tapering space section does not allow oil to move to the bearing section regardless of the air migration into the oil surface, and the pressure difference caused by the space ratio naturally pushes air out to cancel the migrated status. The space ratio secures oil at any position.

The invention was described in detail based on examples; however, this invention is not limited to the examples and it is apparent for those skilled in art that a variety of modifications can be performed within the objective. For example, this invention used a system with a fixed shaft to describe the examples, but it is also applicable in the same manner to a rotatable system; this invention is not limited to the usage as a motor and is also applicable to a variety of systems which use bearings other than motors.

What is claimed is:

1. A bearing seal system comprising:

a shaft formed either on a rotating member or a fixed member;

a cylindrical bag section formed on a non-shaft member, said shaft being inserted into said bag;

a pair of radial bearing sections formed in said cylindrical bag section, said rotating member and fixed member being relatively rotatably supported by said pair of radial bearing sections;

a tapering space section being positioned adjacent to at least one of said radial bearing sections formed in an outer shaft direction; and,

oil which is filled from said bag section to said tapering space section; wherein:

(1) a minimum space in said tapering space section is formed at an inner end of said tapering space section on said radial bearing section side and a maximum space in said tapering space section is formed at an outer end of said space tapering section which is on an opposite side of said radial bearing,

(2) a slope angle is defined between said tapering space section and said shaft, where said slope angle of the tapering space section, as viewed from the inner end of said tapering space section to the outer end of said tapering space section, is at least 0° or more;

(3) a space between the outer end of said tapering space section and said shaft is less than 0.8 mm and said slope angle of the tapering space section is 45 or less; and

(4) a capacity of said tapering space section is at least 5% of that of said bag section, at least 100% of that of said radial bearing section, and a distance between the outer end of said tapering space section and said shaft is at least twice that between the inner end of said tapering space section and said shaft.

2. The bearing seal system of claim **1** wherein an oil circulation hole is formed by connecting both ends of said radial bearing in said bag section.

3. A bearing seal system comprising:

a radial bearing section for rotatably supporting a rotating member against a fixed member, said radial bearing section being formed in one of said rotating member and said fixed member;

oil which is filled between said fixed member and said rotating member; and

a tapering space section which is formed in the axial direction on at least one of a top side and a bottom side of said radial bearing section, wherein:

(1) a minimum space in said tapering space section is formed at an inner end of said tapering space section and is on said radial bearing section side and a maximum space of the tapering space section is formed at an outer end of said tapering space section and is on an opposite side from said radial bearing section;

(2) a slope angle is defined between said tapering space section of the radial bearing section and an opposite wall, where said slope angle as viewed from the inner end of said tapering space section to the outer end of said tapering space section, is at least 0°;

(3) a distance between the outer end of said tapering space section and said opposite wall is 0.8 mm or less and the slope angle of said tapering space section is 45° or less; and

(4) a capacity of said tapering space section is at least 100% that of said radial bearing and the distance between the outer end of said tapering space section to said opposite wall is at least twice that from the inner end of said tapering space section to said opposite wall.

4. The bearing seal system of claim **3** wherein an oil circulation hole which communicates the inner end of said tapering space section with said radial bearing is formed.

5. The bearing seal system of claims **1** or **3** wherein a groove is extended from said tapering space section to its outer side in the axial direction.

6. A bearing seal system including:

a bearing comprising a radial bearing and a thrust bearing to rotatably support a rotating member against a fixed member,

a tapering space section provided on a top side and a bottom side of said radial bearing, and

oil which is filled from said tapering space section provided on said bottom side of said radial bearing to said tapering space section provided on said top side of radial bearing wherein:

(1) a minimum space in each of said tapering space sections is formed at an inner end of the tapering space section and a maximum space in each of said tapering space sections is formed at an outer end of said tapering space section

(2) a slope angle is define between each of said tapering space sections of said tapering space section each tapering space section, as viewed from the inner end to the outer end of said tapering space section, is at least 0°,

(3) a distance between the outer end of each tapering space section and the opposite wall is 0.8 mm or less and the slope angle of each tapering space section is 45° or less, and

(4) a capacities of both capacity of each tapering space section is at least 10% of of a capacity of a space between the two tapering space sections, and at least 100% of a capacity of the radial bearing, and the distance between the outer end of each tapering space section and opposite wall is double or more of that between the inner end of each tapering space and opposite wall.

7. The bearing seal system of claim 6 wherein an oil circulation hole to allow each tapering space section to communicate with each other is formed on an outer side of said bearing.

8. The bearing seal system of claims 1, 3 or 6 wherein a magnetic fluid is filled inside each tapering space section and a magnetic circuit is formed in that magnetic flux is strong at the inner end of each tapering space section and weak at the outer end of each tapering space section; the magnetic field of density gradient magnetic flux circuitry covering at least half of each tapering space section at a predetermined level and in a predetermined direction.

9. The bearing seal system of claims 1, 3 or 6 wherein more than ⅔ of the space of each tapering space section along an axial direction is formed as parallel space of dimension 0.4 mm or more.

10. A bearing seal comprising:

a thrust bearing which relatively rotatably supports a fixed member and a rotating member, said thrust bearing having a cylindrical groove extended in a radial direction and a thrust disk relatively rotatably inserted against said groove;

a hydrodynamic pressure generating groove curved on at least one of four surfaces defined by said thrust bearing, said four surfaces including an upper surface of said cylindrical groove, a bottom surface of said cylindrical groove, an upper surface of said thrust disk, and a bottom surface of said thrust disk;

two a tapering space sections which are formed on an inner radius side of two spaces defined by said four surfaces of said thrust bearing;

oil which is filled to the other tapering space section; and,

a hole which is formed in said thrust disk in an axial direction to communicate with said tapering space sections formed on the inner radius side of said two spaces wherein:

(1) minimum space in each tapering space section is formed at an inner end of said tapering space section on said bearing section side and maximum space in each tapering space section is formed at an outer end of said tapering space section on an opposite side of said bearing section;

(2) a slope angle is defined between each tapering space section and an opposing surface, where said slope angle of each tapering space section, as viewed from the inner end of the tapering space section to the outer end of said tapering space section, is at least 0°;

(3) a distance between the outer end of each tapering space section and said opposing surface is 0.8 mm or less and the slope angle of each tapering space section is 45° or less;

(4) a total clearance in the axial direction in said thrust bearing is 200 μm or less; a total capacity of each tapering space section is at least 100% of a total capacity of said thrust bearing; the total capacity of each tapering space section is at least 30% of a total capacity from one of said tapering space sections to the other tapering space section, and a distance between the outer end of each tapering space section to the opposing surfaces is at least three times that from the inner end of said tapering space section to the opposing surface.

11. The bearing seal system of claim 10 wherein ⅔ or more of said tapering space section remains within the parallel space distance with 0.4 mm or less.

12. The bearing seal system of claims 6 or 10 wherein the hydrodynamic pressure generating groove is comprised to obtain the hydrodynamic pressure of the direction to cancel the centrifugal force which works on oil is canceled.

13. The bearing seal system of claims 1, 3, 6 or 10 wherein the oil quantity during the steady state is set between 0.1 A and 0.9 A position from the inner end of each tapering space section, when the capacity of each tapering space section is A.

14. The bearing seal system of claims 1, 3, 6 or 10 wherein the contact angle between said rotating or fixed member in each tapering space section and oil is at least 15°.

15. The bearing seal system of claim 14 wherein the difference in contact angle between said rotating or fixed member and oil is 15° or less.

16. The bearing seal system of claim 14, wherein the internal wall surface of each tapering space section is made of low-surface tension plastic material.

17. The bearing seal system of claim 14 wherein the surface roughness of the internal surface of each tapering space section is Ra 0.25 μm or less.

18. The bearing seal system of claims 1, 3, 6 or 10 wherein each tapering space section opens at 45° measuring from the inner end to the outer end of each tapering space section.

19. The bearing seal system of claim 18 wherein the average slope angle of each tapering space section is at least 10°.

20. The bearing seal system of claim 18 wherein the slope angle of the tapering space section is predetermined and its cross section of internal wall surface is constructed in straight lines.

21. The bearing seal system of claims 1, 3, 6 or 10 wherein the outer end of the hydrodynamic pressure generating groove formed on said bearing is extended to the inner end of each tapering space section.

* * * * *

EXHIBIT B
TO COMPLAINT

US006343877B1

(12) **United States Patent**
Miura et al.

(10) **Patent No.: US 6,343,877 B1**
(45) **Date of Patent: Feb. 5, 2002**

(54) **SPINDLE MOTOR**

(75) Inventors: **Kazushi Miura**; **Masato Gomyo**; **Takayuki Narita**; **Tokio Tago**; **Takehiko Yazawa**, all of Nagano (JP)

(73) Assignee: **Kabushiki Kaisha Sankyo Seiki Seisakusho**, Nagano (JP)

(*) Notice: Subject to any disclaimer, the term of this patent is extended or adjusted under 35 U.S.C. 154(b) by 0 days.

(21) Appl. No.: **09/549,698**

(22) Filed: **Apr. 14, 2000**

(30) **Foreign Application Priority Data**

Apr. 15, 1999 (JP) 11-107724
Apr. 28, 1999 (JP) 11-123056

(51) **Int. Cl.**[7] **F16C 32/06**
(52) **U.S. Cl.** **384/100**; 384/107
(58) **Field of Search** 384/100, 107, 384/114; 310/90; 360/97.03–99.12

(56) **References Cited**

U.S. PATENT DOCUMENTS

1,998,888 A * 4/1935 Wallgren 384/277
4,999,724 A * 3/1991 McAllister et al. 360/98.08
5,337,374 A * 8/1994 Konishikawa 384/107
5,844,748 A * 12/1998 Dunfield et al. 360/99.08
5,880,545 A * 3/1999 Takemura et al. 310/90
6,072,660 A * 6/2000 Teshima 360/99.08

FOREIGN PATENT DOCUMENTS

JP 8-4769 1/1996

* cited by examiner

Primary Examiner—Thomas R. Hannon
(74) *Attorney, Agent, or Firm*—Sughrue Mion, PLLC

(57) **ABSTRACT**

A potential-difference alleviating member for alleviating and lowering the potential difference, which is an energy difference between a rotating or fixed bearing member and a rotary hub or a fixing frame which are formed of metals of different types, is interposed between the two members so as to prevent the occurrence or advance of potential difference corrosion. Relief portions are respectively provided at a joining interface between a rotary shaft and a thrust plate and a joining interface between a bearing member and the counter plate, and the respective members are welded in the relief portions so as to be integrated.

16 Claims, 6 Drawing Sheets

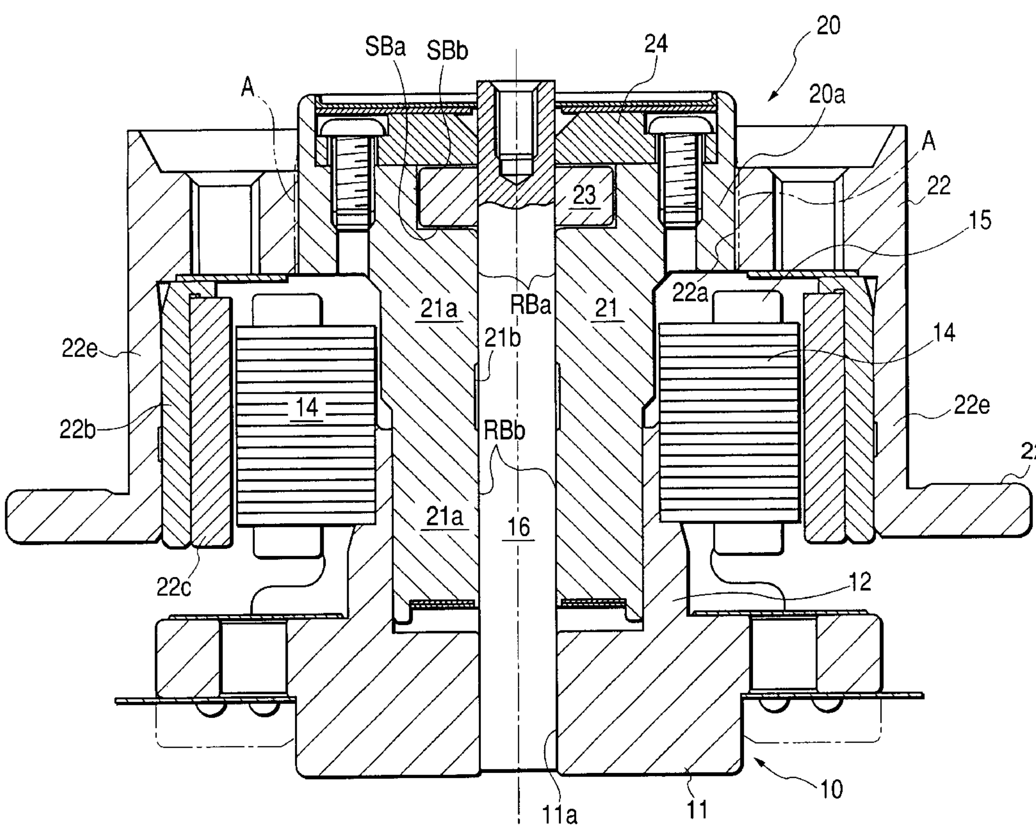


FIG. 1

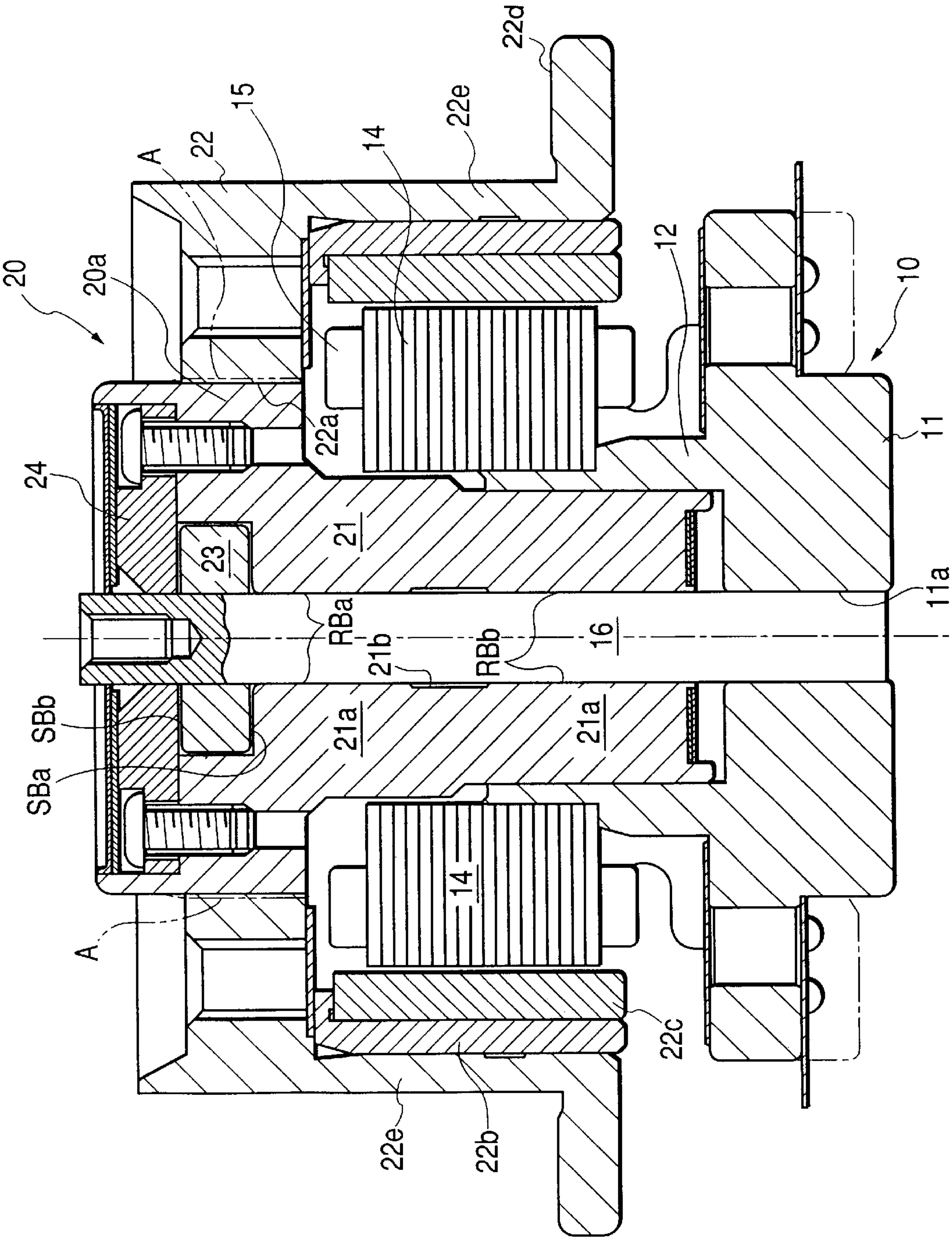
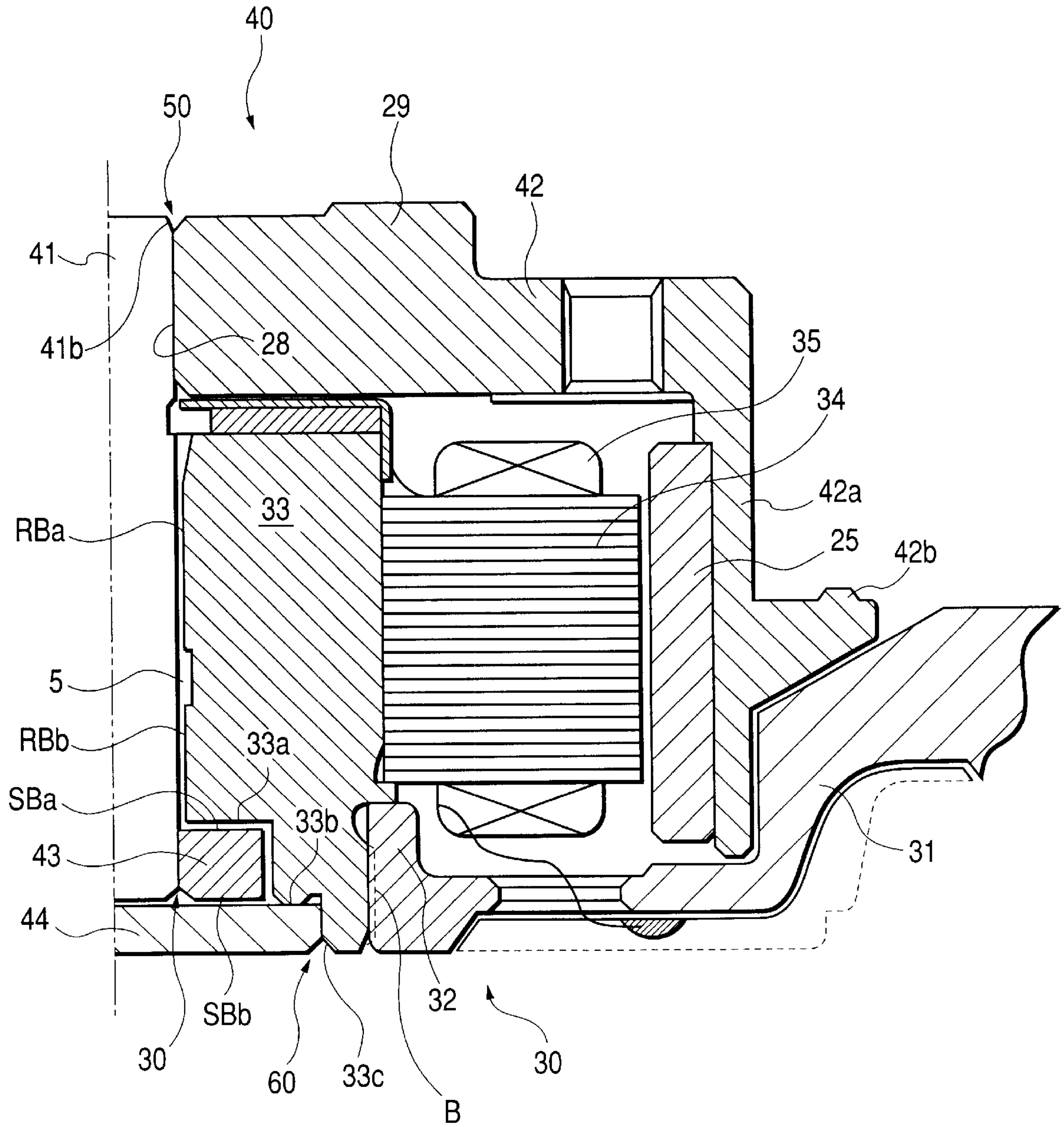

20
20a
A
22
15
14
22e
22d
12
10
11
11a
16
RBb
21b
RBa
21a
21
23
24
SBb
SBa
22a
22c
22b

FIG. 2



40
50
29
42
41
41b
28
35
34
42a
RBa
33
25
42b
5
RBb
33a
SBa
33b
43
31
44
32
SBb
30
60
33c
B
30

FIG. 3

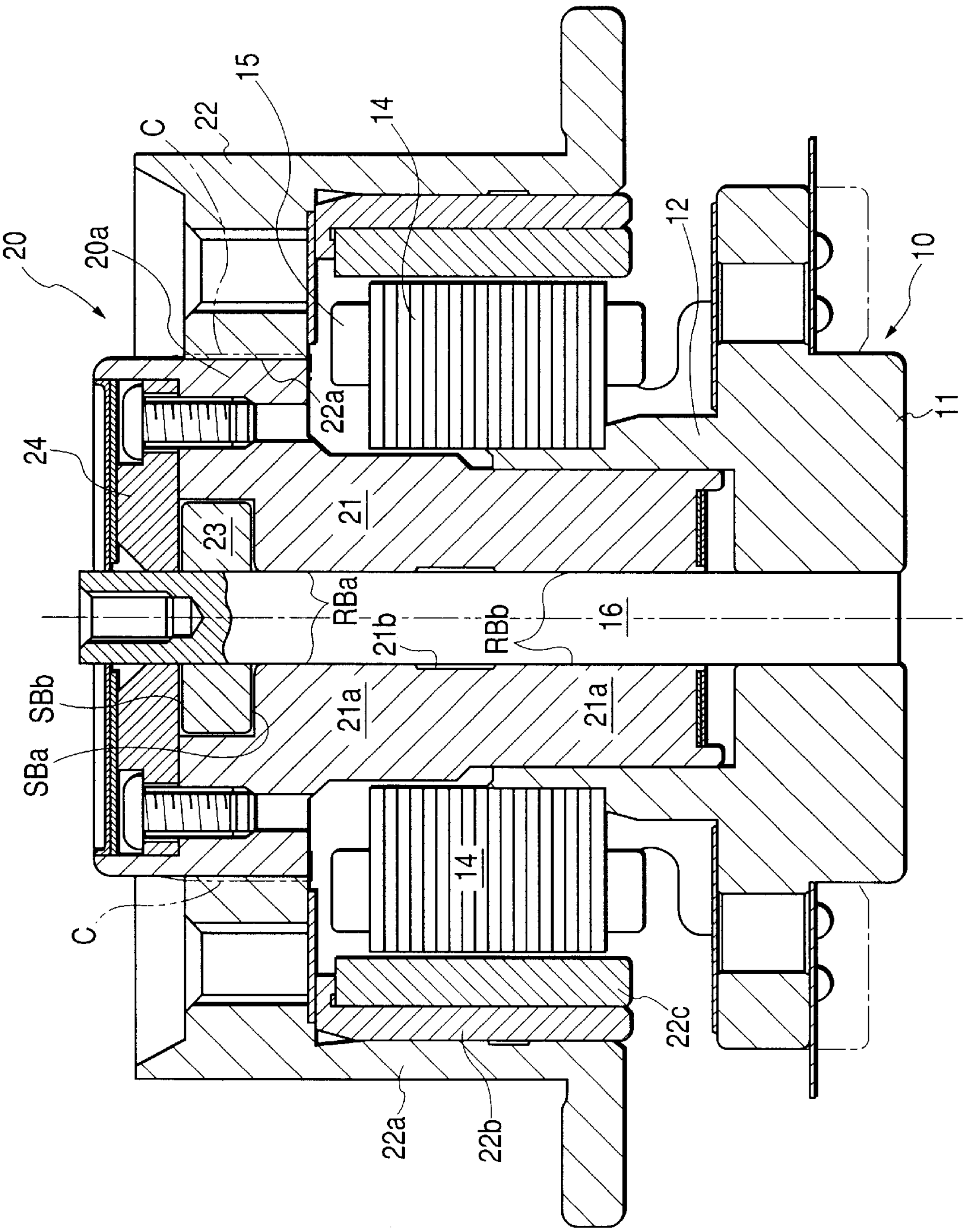

20
20a
C
22
15
14
12
10
11
22a
24
21
23
RBa
21b
RBb
16
SBb
SBa
21a
21a
14
C
22c
22a
22b

FIG. 4A

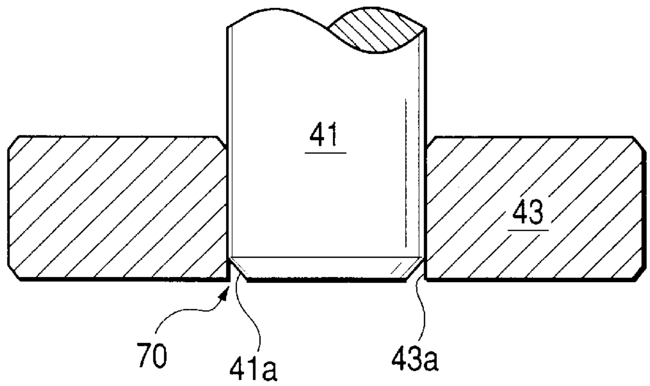


FIG. 4B

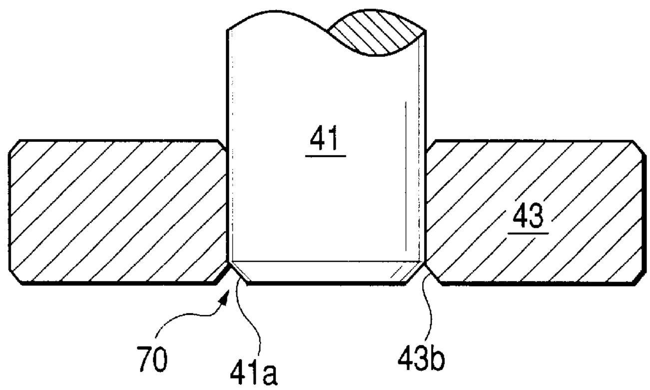


FIG. 4C

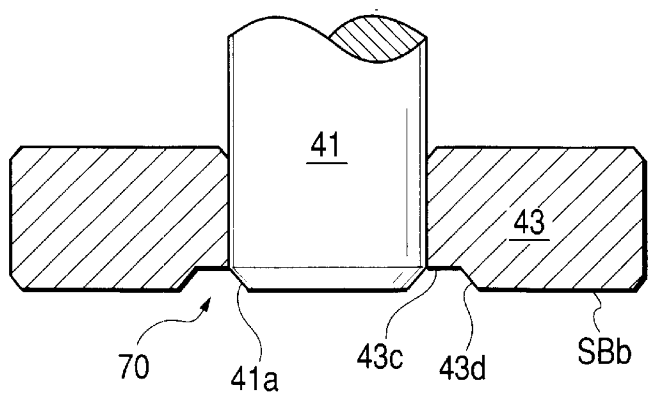


FIG. 5

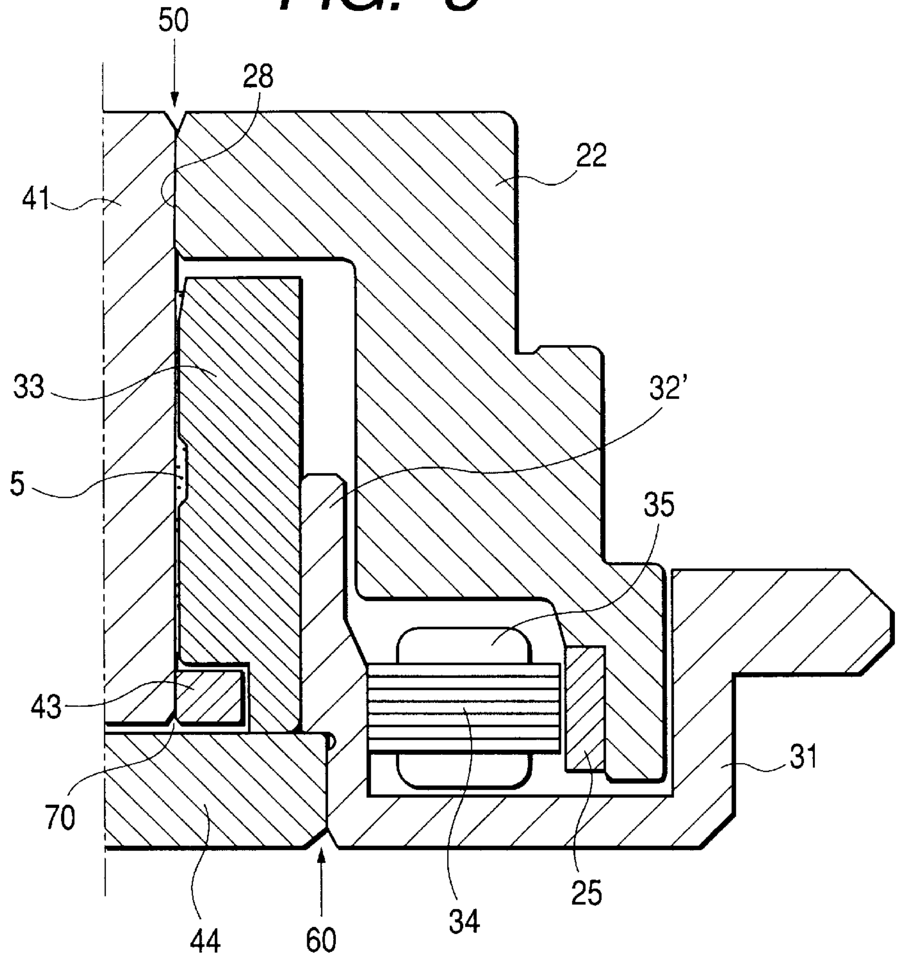


FIG. 6

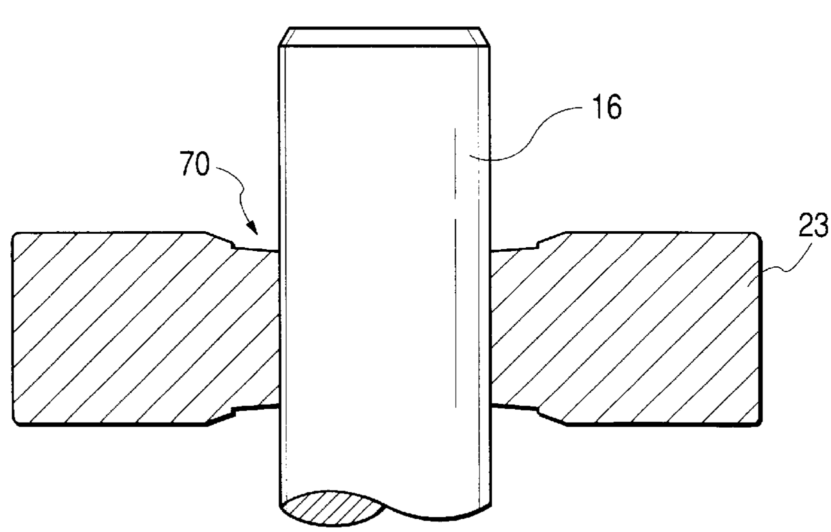
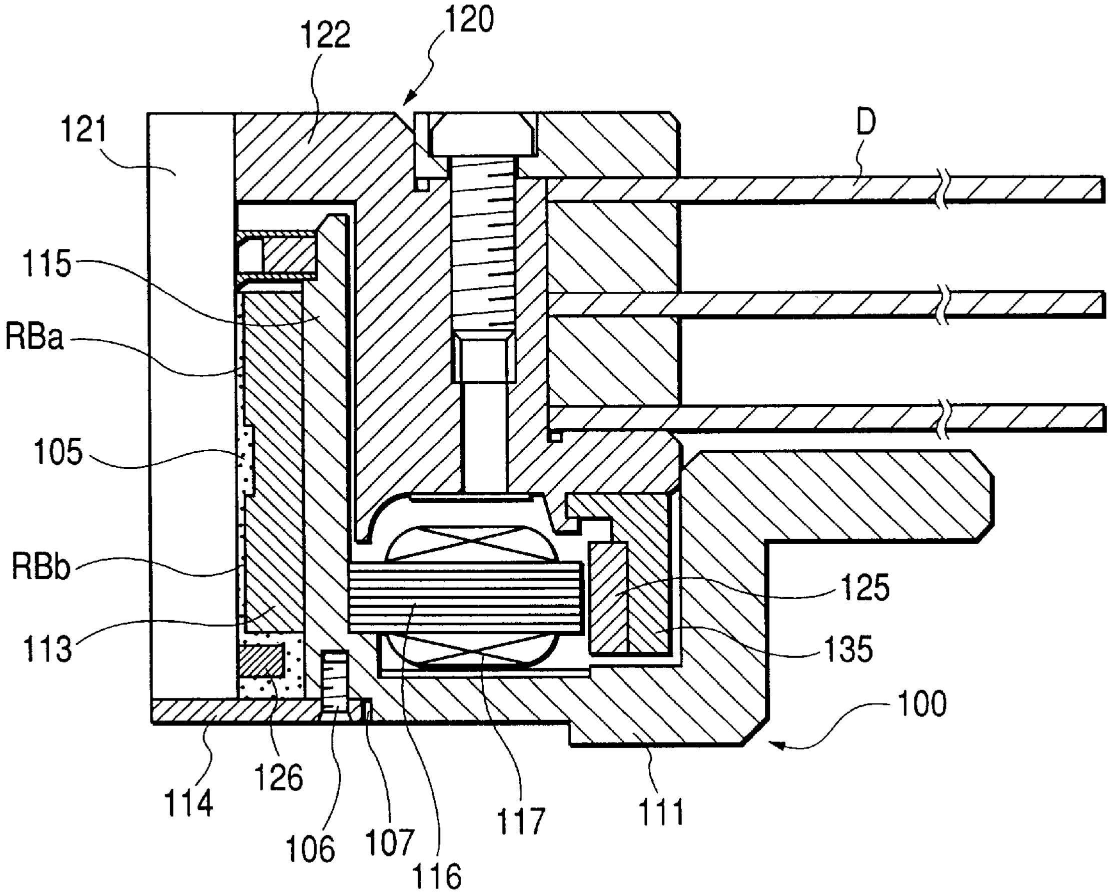


FIG. 7 PRIOR ART



SPINDLE MOTOR

BACKGROUND OF THE INVENTION

The present invention relates to a spindle motor used as an apparatus for rotatively driving a hard disk or the like.

A spindle motor disclosed in, for example, Japanese Patent Publication No. 8-4769A is known as a spindle motor used as an apparatus for rotatively driving a recording medium such as a hard disk. As shown in FIG. **7**, this spindle motor is mainly comprised of a stator assembly **100** and a rotor assembly **120** having driving magnets **125**. The rotor assembly **120** has a hub **122** secured to an upper end portion of a rotary shaft **121** by means of press-fitting, shrinkage fitting, or the like. Meanwhile, the stator assembly **100** has stator cores **116** each formed by winding a coil **117** around a respective salient pole portion. These stator cores **116** are fitted to an outer peripheral portion of a bearing holder **115**.

A bearing sleeve **113** is fitted in an inner peripheral portion of the bearing holder **115**. Radial bearing portions RBa and RBb serving as bearing surfaces for generating hydrodynamic pressure are formed on an inner peripheral surface of the bearing sleeve **113** in such a manner as to be spaced apart from each other in the axial direction. A lubricating fluid **105** such as oil undergoes a pressure rise due to the pumping action of dynamic pressure generating grooves (not shown) when the rotary shaft **121** rotates, and the rotary shaft **121** and the hub **122** are pivotally supported by the hydrodynamic pressure generated by the lubricating fluid **105**.

Further, a thrust plate **126** constituting a thrust hydrodynamic bearing portion is press-fitted and secured to the rotary shaft **121**. Further, a counter plate **114** is fixed at an open end of the bearing holder **115** of a frame **111** through a mechanical coupling means such as fixing screws **106**. The thrust plate **126** is placed between a lower end face of the bearing sleeve **113** and an inner bottom surface of the counter plate **114**, and as the lubricating fluid **105** is present in this space, the rotary shaft **121** is stably supported in the thrust direction by the hydrodynamic pressure generated by the lubricating fluid **105**.

In recent years, a trend toward compact and thin spindle motors for rotatively driving recording medium disks are rapidly underway. In conjunction with this trend, the bearing member (bearing sleeve **113**) supporting the shaft **121** is formed of a metallic material different from the metallic material composing the fixing frame **111**. One reason for this is that a metal excelling in workability is adopted as the metallic material composing the bearing sleeve **113** so that the inside-diameter portion of the bearing sleeve **113** can be machined satisfactorily. In this case, the bearing sleeve **113** formed of a different type of metallic material is integrally joined to the fixing frame **111** by means of press-fitting, shrinkage fitting, or the like.

In a spindle motor in which different types of metallic material are integrally joined together, if an electrolyte having a large dielectric constant, such as water, penetrates the joint, a local battery is formed between these metallic materials of different types, and anodic dissolution occurs due to the local battery, resulting in the so-called potential difference corrosion. The portion where such potential difference corrosion occurs is scattered in due course of time in the form of dust, and causes damage to the recording medium disk or the magnetic head. Accordingly, in the case of an apparatus for which cleanliness is required, such as a hard disk drive (HDD), it is desirable to reliably prevent the occurrence of the aforementioned potential difference corrosion.

In recent years when motors are required to be thinner, it has become impossible to secure a sufficient joining length in the joining of the rotary shaft and the thrust plate and in the joining of the rotary shaft and the hub. Consequently, there have arisen problems in that it is difficult to obtain desired shock-resisting performance (e.g., 1,0000 G or more) and joining strength capable of withstanding an external stress during assembly, thereby making it difficult to produce a thin motor.

For instance, in FIG. **7**, various joining methods are adopted in joining the counter plate **114** and the frame **111** or in joining the counter plate **114** and the bearing sleeve **113**. In a case where the fixing screws **106** shown in FIG. **7** are used to effect fastening, the heads of the fixing screws **106** hinder the attempt to produce a thin motor. In a case where the counter plate **114** is fixed by a calking method, the calked portion must be made to project from the bottom surface of the counter plate **114**, which also hinders the attempt to produce a thin motor. Further, in a case where the counter plate **114** is fixed by a press-fitting method, since a sufficient joining length cannot be obtained, the joining strength lacks.

SUMMARY OF THE INVENTION

A primary object of the invention is to provide a spindle motor which makes it possible to prevent by a simple arrangement the potential difference corrosion between a bearing member and another member which are formed of metallic materials of different types.

A secondary object of the invention is to provide a spindle motor which can be made thin by increasing the joining strength even in the case of a part whose joining length is short.

In accordance with the invention, the arrangement is provided such that a potential-difference alleviating member for alleviating and lowering the potential difference, which is an energy difference between a rotating or fixed bearing member and a rotary hub or a fixing frame which are formed of metals of different types, is interposed between the two members so as to prevent the occurrence or advance of potential difference corrosion. Accordingly, the working environment of an apparatus such as a hard disk drive (HDD), in particular, for which cleanliness is required, can be made favorable, and the reliability of the apparatus can be improved.

Further, in accordance with the invention, the arrangement is provided such that an insulating resin coating film or a passivation film is interposed between a rotating or fixed bearing member and a rotary hub or a fixing frame which are formed of metals of different types, so as to prevent the occurrence of a local battery and prevent the occurrence or advance of potential difference corrosion. Accordingly, the working environment of an apparatus such as a hard disk drive (HDD), in particular, for which cleanliness is required, can be made favorable, and the reliability of the apparatus can be improved.

Furthermore, in accordance with the invention, the arrangement is provided such that relief portions are respectively provided at a joining interface between the rotary shaft and the thrust plate and a joining interface between the bearing member and the counter plate or a joining interface between the fixing frame and the counter plate, and the respective members are welded in the relief portions so as to be integrated. Accordingly, even if the joining length of the members is relatively short, it is possible to obtain a sufficient joining strength and improve the shock resistance of

the motor itself. As a result, the perpendicularity of the thrust plate with respect to the rotary shaft, for example, can be maintained stably, and the reliability of the motor can be improved.

BRIEF DESCRIPTION OF THE DRAWINGS

In the accompanying drawings:

FIG. **1** is an explanatory cross-sectional view showing a hard-disk driving motor of a shaft fixed type according to a first embodiment of the present invention;

FIG. **2** is an explanatory half cross-sectional view showing a hard-disk driving motor of a shaft rotating type according to a second embodiment of the present invention;

FIG. **3** is an explanatory cross-sectional view showing a hard-disk driving motor of a shaft fixed type according to a third embodiment of the present invention;

FIGS. **4**A to **4**C are cross-sectional views showing the structure for joining a rotary shaft and a thrust plate;

FIG. **5** is a half cross-sectional view showing a spindle motor according to a fourth embodiment of the present invention;

FIG. **6** is a cross-sectional view showing the structure for joining the fixed shaft and the thrust plate shown in FIG. **1**; and

FIG. **7** is a half cross-sectional view showing a related spindle motor.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

Hereafter, a description will be given of the embodiments of the invention. First, referring to the drawings, a description will be given of the structure of a hard disk drive (HDD) to which the invention is applied.

The HDD spindle motor of a shaft fixed type, which is a first embodiment of the present invention, shown in FIG. **1** is comprised of a stator assembly **10** serving as a fixed member and a rotor assembly **20** serving as a rotating member which is assembled to the stator assembly **10** from an upper side thereof in the drawing. Of these assemblies, the stator assembly **10** has a fixing frame **11** which is screwed down to an unillustrated fixed base. A hollow cylindrical bearing holder **12** is formed on a substantially central portion of the fixing frame **11** in such a manner as to be integrally provided uprightly, and stator cores **14** are fitted to an outer peripheral surface of the bearing holder **12**. Driving coils **15** are respectively wound around salient pole portions of the stator cores **14**.

A fixed shaft **16** formed of a stainless steel (SUS 420J2; indication based on JIS) is fixed in a shaft-fixing hole **11***a* of the fixing frame **11** in such a manner as to project upwardly. This fixed shaft **16** is disposed concentrically with the bearing holder **12**, and an upper end portion of the fixed shaft **16** is also screwed down to the unillustrated fixed base. A bearing sleeve **21** serving as a rotating-shaft bearing member making up a part of the rotor assembly **20** is rotatably fitted on an outer periphery of the fixed shaft **16**, and a rotary hub **22** for mounting an unillustrated recording medium such as a magnetic disk is joined to an outer periphery of the bearing sleeve **21**.

A cylindrical large-diameter portion **20***a* for joining, which is formed in such a manner as to project outwardly in the radial direction, is disposed in an upper end portion of the bearing sleeve **21**. A joining hole **22***a*, which is formed penetratingly in a central portion of the rotary hub **22**, is integrally joined to an outer peripheral surface of the large-diameter portion **20***a* for joining by means of press-fitting or shrinkage fitting. The rotary hub **22** is formed of an aluminum group material for the purpose of light weight, and has a cylindrical body **22***e*. Annular driving magnets **22***c* are attached to an outer periphery of the cylindrical body **22***e* with a back yoke **22***b* placed therebetween. These magnets **22***c* are disposed in such a manner as to annularly oppose outer peripheral-side end faces of the stator cores **14** in close proximity thereto. Further, the cylindrical body **22***e* has a disk-mounting surface **22***d* for mounting the recording medium disk on its outer peripheral portion.

Meanwhile, the bearing sleeve **21** is formed of a copper group material or a stainless steel metal to facilitate drilling and the like. A pair of bearing projections **21***a* serving as a pair of radial bearings are formed on an inner periphery of a central hole, which is provided in the bearing sleeve **21**, in such a manner as to be axially spaced apart a predetermined distance. Further, hydrodynamic surfaces formed on inner peripheral surfaces of these bearing projections **21***a* are disposed in such a manner as to proximately oppose hydrodynamic surfaces formed on an outer peripheral surface of the fixed shaft **16**, thereby forming a pair of radial hydrodynamic bearing portions RBa and RBb which are adjacent to each other in the axial direction. More specifically, the hydrodynamic surface on the bearing sleeve **21** side and the hydrodynamic surface on the fixed shaft **16** side in each of the pair of radial hydrodynamic bearing portions RBa and RBb are opposingly disposed circumferentially with a very small gap of several microns therebetween. A lubricating fluid such as oil, a magnetic fluid, or air is charged in the bearing space having the very small gap in such a manner as to continue in the axial direction. It should be noted that oil is used as the lubricating fluid in this embodiment.

A fluid storage portion **21***b*, which is formed by radially recessing the inner peripheral portion of the central hole in the bearing sleeve **21**, is formed between the pair of radial hydrodynamic bearing portions RBa and RBb, and a sufficient quantity of lubricating fluid is stored in the fluid storage portion **21***b*.

At least one of the hydrodynamic surfaces of the bearing sleeve **21** and the fixed shaft **16** is annually recessed to form unillustrated radial dynamic pressure generating grooves of, for example, herringbone shape in such a manner as to be axially divided into two blocks. Thus the lubricating fluid is pressurized by the pumping action of the radial dynamic pressure generating grooves during rotation to generate hydrodynamic pressure, and the rotary hub **22** is pivotally supported in the radial direction by the hydrodynamic pressure of the lubricating fluid.

In the axially opposite end portions of the bearing space which form the radial hydrodynamic bearing portions RBa and RBb, a pair of capillary seal portions are respectively disposed in such a manner as to axially sandwich the radial hydrodynamic bearing portions RBa and RBb. Each of these capillary seal portions is formed by gradually enlarging the gap between the bearing sleeve **21** and the fixed shaft **16** in the radially outward direction in a tapered manner by an inclined surface formed on the bearing sleeve **21**. The dimension of the gap of the capillary seal portion disposed on the inward side of the bearing is set to a range of 20 μm to 300 μm, for example. These capillary seal portions are so arranged that the level of the lubricating fluid is located there when the motor either rotates or is at a standstill.

A disk-shaped thrust plate **23** is secured to an illustrated upper end portion of the fixed shaft **16**. This thrust plate **23**

is disposed so as to be accommodated in a hollow cylindrical recessed portion formed in a central portion of the upper end of the bearing sleeve **21**. Axially proximately opposing surfaces of the thrust plate **23** and the bearing sleeve **21** in the recessed portion of the bearing sleeve **21** are formed as hydrodynamic surfaces, thereby forming a lower thrust hydrodynamic bearing portion SBa.

Further, a counter plate **24** formed of a large disk-shaped member is secured to the upper end portion of the bearing sleeve **21** so as to be located in close proximity to the illustrated upper hydrodynamic surface of the thrust plate **23**. An upper thrust hydrodynamic bearing portion SBa is formed by the hydrodynamic surface provided on the lower surface of the counter plate **24** and the hydrodynamic surface of the thrust plate **23** side.

Both hydrodynamic surfaces on the thrust plate **23** side in the pair of thrust hydrodynamic bearing portions SBa and SBb which are disposed axially adjacent to each other, and both hydrodynamic surfaces on the bearing sleeve **21** and the counter plate **24** side which are respectively opposed thereto, are disposed in face-to-face relation to each other in the axial direction with very small gaps of several microns therebetween. The lubricating fluid such as oil, a magnetic fluid, or air is charged in the bearing spaces having the very small gaps in such a manner as to continue in the axial direction through outer peripheral-side passages in the thrust plate **23**.

Further, at least one of the hydrodynamic surfaces of the thrust plate **23** on the one hand, and the hydrodynamic surfaces of the bearing sleeve **21** and the fixed shaft **16** on the other hand, is annually recessed to form unillustrated thrust dynamic pressure generating grooves of, for example, herringbone shape in such a manner as to be radially divided into two blocks. Thus the lubricating fluid is pressurized by the pumping action of the thrust dynamic pressure generating grooves during rotation to generate hydrodynamic pressure, and the rotary hub **22** is pivotally supported in the thrust direction by the hydrodynamic pressure of the lubricating fluid.

Next, a description will be given of the technique of the invention for preventing the potential difference corrosion which can occur between the bearing member (bearing sleeve) and another member which are formed of metallic materials of different types.

As described above, the bearing sleeve **21** is formed of a copper group material, e.g., phosphor bronze which is one of copper alloys, while the rotary hub **22** which is integrally joined to the bearing sleeve **21** is formed of an aluminum group material, e.g., an aluminum material. These metals of different types are joined, thereby forming a one-piece rotating member. A potential-difference alleviating member A is interposed between the joined surfaces of the bearing sleeve **21** and the rotary hub **22**. This potential-difference alleviating member A is formed of a metallic material, such as a nickel material, whose ionization tendency in the electrochemical series with respect to solution, e.g., plain water (tap water), is positioned between that of copper and that of aluminum. This potential-difference alleviating member A is formed on at least one of the joined surfaces of the bearing sleeve **21** and the rotary hub **22** in film form by plating processing, vapor deposition processing, or coating.

The ionization tendency refers to the tendency whereby a metal produces cations when coming into contact with a liquid, particularly water, and can be quantitatively evaluated by the standard electrode potential of the metal. The list of metals in which their ionization tendencies with respect to solution are arranged in the order of their magnitude is referred to as the electrochemical series.

In a case where the metallic materials of different types are copper and aluminum, metallic materials whose ionization tendencies in the electrochemical series with respect to plain water are positioned between that of copper and that of aluminum are Co, Mo, Cr, and Ni, while metallic materials whose ionization tendencies in the electrochemical series with respect to saline water are positioned therebetween are Fe, Sn, Co, W, Cr, Mo, and Ni. The ionization tendencies in the electrochemical series with respect to solution types are thus known. Accordingly, the material which is used as the potential-difference alleviating member A is selected on the basis of the aqueous solution which is considered to attach to the metals as well as the two metallic materials to be joined.

In the above-described embodiment, supposing that the potential-difference alleviating member A is not provided, since the rotating member is used in which the bearing sleeve **21** formed of a copper material and the rotary hub **22** formed of an aluminum material are joined, if an electrolyte having a large dielectric constant, such as water, penetrates the joint, a local battery is formed between the metallic materials of different types. Hence, anodic dissolution can possibly occur due to the local battery, resulting in potential difference corrosion. In contrast, in accordance with the invention, since the nickel film is provided as the potential-difference alleviating member A between the bearing sleeve **21** and the rotary hub **22**, the potential difference between the two members **21** and **22** becomes small due to the potential-difference alleviating member A interposed between the two members **21** and **22**, thereby making it possible to prevent the generation of the local battery and hamper the occurrence or advance of the potential difference corrosion. This action of preventing the potential difference corrosion is effective when one component part is formed by joining different types of metals as in the case of the bearing sleeve **21** and the rotary hub **22**, and an energy difference (potential difference) occurs between the joined members.

As described above, the metal whose ionization tendency in the electrochemical series with respect to solution is positioned between those of two metals to be joined, i.e., the potential-difference alleviating member A, can be selected from among a number of materials. Hence, it suffices to select a material to be formed by taking into consideration a desired manufacturing method such as plating processing, vapor deposition processing, or coating. If the material is selected from this perspective, by merely adding such as a plating process to normal machining and assembling processes, it becomes possible to easily provide the potential-difference alleviating member A having a satisfactory function.

Although, in the first embodiment, a description has been given of a case in which a component part formed by joining metals of different types is formed by a copper material including a copper alloy and an aluminum material including an aluminum alloy, the selection of these metals of different types may be changed as required.

Meanwhile, the invention is similarly applicable to a spindle motor of a shaft rotating type whose half cross sectional view is shown in FIG. **2**, which is a second embodiment of the present invention.

The overall HDD spindle motor of the shaft rotating type shown in FIG. **2** is comprised of a stator assembly **30** serving as a fixed member and a rotor assembly **40** serving as a rotating member assembled to the stator assembly **30** from

an upper side thereof in the drawing. Of these assemblies, the stator assembly **30** has a fixing frame **31** which is screwed down to an unillustrated fixed base. The fixing frame **31** is formed of an aluminum group material to attain light weight. A bearing sleeve **33** serving as a fixed bearing member formed in a hollow cylindrical shape is integrally joined to the inner side of an annular mounting portion **32**, which is formed in such a manner as to be provided uprightly in a substantially central portion of the fixing frame **31**, by press-fitting or shrinkage fitting.

The lower outer peripheral surface of the bearing sleeve **33** is formed such that its radial dimension substantially coincides with the radial dimension of the outer peripheral surface of the annular mounting portion **32**. Stator cores **34** are fitted to an attaching surface formed by an outer peripheral surface of the bearing sleeve **33**. Driving coils **35** are respectively wound around salient pole portions provided in the stator cores **34**. In the embodiment shown FIG. **2**, although the stator cores **34** are fitted to the attaching surface formed by the outer peripheral surface of the bearing sleeve **33**, an arrangement may be provided such that the annular mounting portion **32** is extended upwardly, and the stator cores **34** are attached to an outer peripheral surface of that annular mounting portion **32**.

A rotary shaft **41** formed of a stainless steel (SUS 420J2) or the like and making up a part of the rotor assembly **40** is rotatably inserted in a central hole provided in the bearing sleeve **33**. Namely, hydrodynamic surfaces formed on the inner peripheral surface of the bearing sleeve **33** are disposed in such a manner as to proximately oppose hydrodynamic surfaces formed on the outer peripheral surface of the rotary shaft **41**, thereby forming the pair of radial hydrodynamic bearing portions RBa and RBb which are adjacent to each other in the axial direction. The hydrodynamic surface on the bearing sleeve **33** side and the hydrodynamic surface on the rotary shaft **41** side in each of the pair of radial hydrodynamic bearing portions RBa and RBb are opposingly disposed circumferentially with a very small gap of several microns therebetween. A lubricating fluid such as oil, a magnetic fluid, or air can be used in the bearing space.

The bearing sleeve **33** is formed of a copper group material or a stainless steel to facilitate machining, and radial dynamic pressure generating grooves of, for example, herringbone shape are formed in its inner periphery in such a manner as to be axially divided into two blocks. Thus a rotary hub **42** together with the rotary shaft **41** is pivotally supported in the radial direction by the hydrodynamic pressure of the lubricating fluid during rotation.

The substantially cup-shaped hub **42** on which a recording medium such as a magnetic disk is mounted is secured to one end of the rotary shaft **41** by means of a joining member which will be described later. The hub **42** has a hollow cylindrical portion **42***a* to which the disk is fitted, as well as a disk mounting surface **42***b* which expands outwardly from the lower end of the hollow cylindrical portion **42***a* for mounting the disk thereon. Annular driving magnets **25** having magnetized poles are fitted to an inner peripheral surface of the hollow cylindrical portion **42***a* of the hub **42**, and inner peripheral surfaces of the driving magnets **25** are opposed to outer peripheral surfaces of the stator cores **34** with an appropriate interval therebetween. Here, since the hub **42** is formed of a magnetic material such as iron, the hub **42** itself can be made to function as a back yoke for the driving magnets **25**. Accordingly, in this embodiment, since the yoke which is a separate component is omitted, as compared with a hub **42** having an identical outside diameter and the yoke, the inner space of the hub **42**, i.e., the space for disposing the armature, can be made large. Accordingly, it is possible to obtain a relatively large motor torque. It should be noted that in a case where the hub **42** is formed of a nonmagnetic material such as an aluminum alloy, a yoke formed of a magnetic material is interposed between the hub **42** and the driving magnets **25**.

Meanwhile, a disk-shaped thrust plate **43** is secured to the other end side, i.e., on the lower side in the drawing, of the rotary shaft **41** by means of a joining member which will be described later. This thrust plate **43** is disposed so as to be accommodated in a recessed portion **33***a* formed in a central portion of the lower end side of the bearing sleeve **33**. The upper thrust hydrodynamic bearing portion SBa is formed by hydrodynamic surfaces formed by axially proximately opposing end faces of the thrust plate **43** and the bearing sleeve **33** in the recessed portion **33***a* of the bearing sleeve **33**.

Further, a disk-shaped counter plate **44** larger than the thrust plate **43** is secured in a lower end-side opening of the bearing sleeve **33** by a joining member, which will be described later, in such a manner as to be located in close proximity to the illustrated upper hydrodynamic surface of the thrust plate **43**. Then, the lower thrust hydrodynamic bearing portion SBb is formed by the hydrodynamic surface provided on an upper end face of the counter plate **44** and the hydrodynamic surface on the thrust plate **44** side.

The hydrodynamic surfaces on the thrust plate **43** side in the pair of thrust hydrodynamic bearing portions SBa and SBb which are disposed axially adjacent to each other, and the hydrodynamic surfaces on the bearing sleeve **33** and the counter plate **44** side which are respectively opposed thereto, are disposed in face-to-face relation to each other in the axial direction with very small gaps of several microns therebetween. A lubricating fluid **5** is charged in the bearing spaces having the very small gaps in such a manner as to continue in the entire axial direction through outer peripheral-side passages in the thrust plate **43**.

At least one of the hydrodynamic surfaces of the thrust plate **43** on the one hand, and the hydrodynamic surfaces of the bearing sleeve **33** and the counter plate **44** on the other hand, is annually recessed in the usual manner to form thrust dynamic pressure generating grooves of herringbone shape or spiral shape. Thus, when the thrust plate **43** is rotated in conjunction with the rotation of the rotor assembly **40**, the rotor assembly **40** including the rotary shaft **41** and the hub **42** is pivotally supported in the thrust direction by the hydrodynamic pressure of the thrust dynamic pressure generating grooves.

As described above, the bearing sleeve **33** is formed of a copper group material, specifically phosphor bronze, to facilitate machining, while the fixing frame **31** which is integrally joined to the bearing sleeve **33** is formed of an aluminum group material, specifically an aluminum material. These metals of different types are joined. A potential-difference alleviating member B is interposed between the joined surfaces of the bearing sleeve **33** and the fixing frame **31**. In the same way as in the already-described embodiment shown in FIG. **1**, this potential-difference alleviating member B is formed of a metallic material, such as a nickel material, whose ionization tendency in the electrochemical series is positioned between that of a copper group material and that of an aluminum group material. This potential-difference alleviating member B can be formed by being coated on at least one of the joined surfaces of the bearing sleeve **33** and the fixing frame **31** in film form by plating processing, vapor deposition processing, or coating.

It should be noted that the potential-difference alleviating member B may be formed by a passivation film B coated on at least one of the joined surfaces of the bearing sleeve **33** and the fixing frame **31**.

This passivation film B is an oxide film excelling in corrosion resistance, and can be obtained by subjecting the joined surface of the bearing sleeve **33** or the fixing frame **31** to electroless nickel-phosphor plating and by oxidizing and giving passivity to the plated film by being left to stand for a predetermined duration.

It should be note that, as for the passivation film B, the metallic material itself forming the bearing sleeve **33** or the fixing frame **31** may be used as the passivation film instead of using a plating material different from the metal to be joined. For example, an alumite film may be formed on the joined surface of the fixing frame **31** which is formed of an aluminum material and is joined to the bearing sleeve **33**, and it is possible to prevent the formation of a local battery in the event that an electrolyte having a large dielectric constant, such as water, has penetrated the joined portions of the fixing frame **31** and the bearing sleeve **33**.

In this embodiment as well, the energy difference between the bearing sleeve **33** and the fixing frame **31** in which metals of different types are joined, i.e., the potential difference between the two members **33** and **31**, can be alleviated and lowered by the potential-difference alleviating member B interposed between the two members **33** and **31**, thereby making it possible to prevent the occurrence or advance of potential difference corrosion.

Next, in a third embodiment shown in FIG. **3**, instead of the potential-difference alleviating member A in the first embodiment shown in FIG. **1**, an insulating resin coating film C is interposed between the joined surfaces of the bearing sleeve **21** and the rotary hub **22**. This resin coating film C is continuously formed over the entire circumferential periphery ranging from the inner joined portions of the bearing sleeve **21** and the rotary hub **22**, to which water or the like is liable to be attached, to outer exposed surfaces of the bearing sleeve **21** on the upper and lower sides thereof in the drawing. In the inner joined portions of the bearing sleeve **21** and the rotary hub **22**, a region is provided where the resin coating film C is not formed and is left in a notched state, so that the bearing sleeve **21** and the rotary hub **22** are made electrically conductive. Accordingly, the joined surfaces of the bearing sleeve **21** and the rotary hub **22** are made electrically conductive at the notched portion of the resin coating film C.

In the embodiment having the above-described configuration, since the bearing sleeve **21** and the rotary hub **22** formed of metals of different types are electrically insulated by the resin coating film C, even if waterdrops are attached, the local battery is not formed. Consequently, it is possible to prevent the occurrence or advance of potential difference corrosion. Since the attachment of the waterdrops cause a problem in the joined portions exposed to the outside, in this embodiment in which the resin coating film C is continuously formed up to the outer exposed surfaces of the bearing sleeve **21** extending continuously at the joined surfaces, the formation of the local battery can be prevented satisfactorily even if the electrolyte such as water is attached to the outer exposed surfaces of the joined surfaces.

The arrangement in which the potential-difference alleviating member or the passivation film is formed over the entire periphery up to the outer exposed surfaces of the bearing sleeve **21** extending continuously at the joined surfaces can be also applied to the embodiments already described.

In this embodiment, since the bearing sleeve **21** and the rotary hub **22** are electrically insulated by the resin coating film C, while inside part of the joined surfaces is made electrically conductive without the resin coating C, an arrangement can be provided to ground the rotary hub **22** through that conductive portion. Accordingly, even if static electricity has been generated in the rotary hub **22**, discharging can be effected smoothly, so that damage or the like to the magnetic head due to the static electricity can be prevented.

Such a resin coating film C is similarly applicable to the spindle motor of the shaft rotating type shown in FIG. **2**. If a similar resin coating film C is formed between the bearing sleeve **33** and the fixing frame **31**, it is possible to obtain similar effect and advantages.

It should be noted that the invention can be similarly applied to any portion if it is a portion where metals of different types are joined. For example, in the embodiment shown in FIG. **2**, a potential-difference alleviating member may be interposed between the joining portions of the rotary shaft **41** and the rotary hub **42**.

Next, a description will be given of the technique of the invention for enhancing the joining strength of component parts even if joining length is small.

FIGS. **4**A to **4**C are diagrams explaining the structure for joining the rotary shaft **41** and the thrust plate **43** of the spindle motor in accordance with the second embodiment.

If the spindle motor is made thin and is designed to a height of, for example, 5 mm or thereabouts, the joining length of the rotary shaft **41** and the thrust plate **43** becomes less than 1 mm. Accordingly, the joining strength becomes weak since a sufficient joining length cannot be obtained even if the joining of the two members is effected by the press-fitting method or the shrinkage fitting method. If press-fitting is effected by providing a large press-fitting allowance, there is a possibility of deterioration of the perpendicularity of the thrust plate **43** with respect to the rotary shaft **41**, so that a press-fitting allowance of more than a predetermined amount cannot be provided. Accordingly, in this embodiment, after the rotary shaft **41** and the thrust plate **43** are press-fitted or inserted by an appropriate press-fitting force to such an extent that the deterioration of perpendicularity does not occur, the joining interface portions of the two members are welded together. At this juncture, an axially recessed relief portion **70** is annularly formed in advance at the surface portion of the joining interface portion, and the rotary shaft **41** and the thrust plate **43** are welded in this relief portion **70**.

The shape of the relief portion **70** at the joining interface between the rotary shaft **41** and the thrust plate **43** is formed in one of the shapes shown in FIGS. **4**A, **4**B, and **4**C. Namely, in FIG. **4**A, a tapered surface **41***a* is formed over the entire periphery around the outer peripheral edge of a tip of the rotary shaft **41**, while an inner peripheral surface **43***a* of a central hole of the thrust plate **43** is adjacent to the tapered surface **41***a*. Accordingly, the relief portion **70** of a wedge-shaped cross section is formed, and the two members are welded in this relief portion **70**. It should be noted that the tapered surface **41***a* at the tip of the rotary shaft **41** also functions as a guide portion at the time the thrust plate **43** is press-fitted to the rotary shaft **41**.

FIG. **4**B shows an example in which the tapered surface **41***a* is formed over the entire periphery around the outer peripheral edge of the tip of the rotary shaft **41**, while a tapered surface **43***b* is also formed around the inner peripheral edge of the central hole of the thrust plate **43**. The two members are welded together in this relief portion **70**.

In FIG. 4C, the tapered surface **41***a* is formed over the entire periphery around the outer peripheral edge of the tip of the rotary shaft **41**, while a flat recess **43***c* is formed around the central hole at a bottom surface portion of the thrust plate **43**, a tapered surface **43***d* being formed around its outer periphery. Further, the hydrodynamic surface SBb is formed on its outer side. In the case of this example, a trapezoidal relief portion **70** is formed, and the two members are welded together in this relief portion **70**.

Each of the relief portions **30** formed at the joining interface between the rotary shaft **41** and the thrust plate **43** is formed at a position offset from the region where the dynamic pressure generating grooves are formed in the thrust plate **43**. Accordingly, the dynamic pressure generating grooves are not subjected to limitations by the relief portion **70**, and it is possible to allow desired thrust hydrodynamic pressure to be demonstrated.

Further, as for the welding position, the entire periphery may be welded, or welding may be effected partially at a plurality of locations, insofar as the welding position or positions are located in the relief portion **70**.

As the welding process, it is possible to adopt a plasma welding process, an arc welding process such as TIG welding, an electron beam welding process typified by laser welding, or the like. In this embodiment, the laser welding process is adopted in Which the basic materials to be joined are welded together by fusing the two materials. In this laser welding process, a laser beam emitted from a laser oscillator is focused by using a plurality of mirrors, and is radiated to the joining interface to join the two members. According to such an electron beam welding process, since a welding rod used in the arc welding process is made unnecessary, the buildup of the basic material in the joined interface portions can be minimized. Further, even if a slight buildup has occurred, since the axially recessed relief portion **70** is provided at the joining interface, the built-up portion is accommodated in the relief portion **70**, and can be prevented from projecting from the hydrodynamic surface toward the counter plate **44** side (see FIG. **2**). Accordingly, it is desirable to set the size of the relief portion **70** by taking the size of the built-up portion into consideration.

If the arrangement is provided such that the built-up portion is accommodated in the relief portion **70**, the built-up portion is prevented from being located excessively close to the counter plate **44**, and when the rotor assembly **40** including the thrust plate **43** is rotated, it is possible to prevent the built-up portion from colliding against the bearing surface of the counter plate **44**. Further, although the joined portions of the rotary shaft **41** and the thrust plate **43** are located in the lubricating fluid **5**, since the two members are joined by welding without using an organic solvent such as an adhesive agent, the catalytic action with respect to the lubricating fluid **5** does not occur, so that the characteristics of the lubricating fluid **5** such as oil do not deteriorate.

Next, a description will be given of the structure for joining the bearing sleeve **33** and the counter plate **44** of the spindle motor shown in FIG. **2**.

The disk-shaped counter plate **44** is secured in the opening at the lower end of the bearing sleeve **33** formed in a hollow cylindrical shape. The counter plate **44** has its outer peripheral surface press-fitted to the bearing sleeve **33** with an appropriate press-fitting force, and an outer peripheral edge of its upper end face abuts against a stepped portion **33***b* of the bearing sleeve **33**. Further, an axially recessed relief portion **60** is formed in the portions of the obverse (lower) sides of the joining interface portions of the bearing sleeve **33** and the counter plate **44**, and the two members are integrated by welding in the relief portion **60**. As the welding process, in the same way as the above-described process of joining the rotary shaft **41** and the thrust plate **43**, it is possible to use an electron beam welding process typified by laser welding. Accordingly, at least one of the bearing sleeve **33** and the counter plate **44** is fused by being irradiated with an electron beam, thereby joining the two members.

Further, the shape of the relief portion **60** may be wedge-shaped, triangular, trapezoidal, or other cross-sectional shapes in the same way as the shape of the stepped surface of the relief portion **70** formed at the joining interface between the rotary shaft **41** and the thrust plate **43** shown in FIGS. **4**A to **4**C. It should be noted that a tapered guide portion **33***c* should preferably be formed at an inner peripheral edge of the opening of the bearing sleeve **33** so as to facilitate the press-fitting or insertion of the counter plate **44**. Further, as for the welding position, it is preferable to weld the entire periphery so as to seal the opening.

In the structure for joining the bearing sleeve **33** and the counter plate **44**, the relief portion **60** is provided which is capable of accommodating the built-up portion formed by joining the joining interface portions, and welding is effected in this relief portion **60** to integrate the two members, as described above. Therefore, even if the built-up portion is formed by joining, the attempt to make the overall motor thin is not hampered. Furthermore, since the bearing sleeve **33** and the counter plate **44** are joined by welding, it is possible to reliably prevent the leakage of the lubricating fluid **5** without using an O-ring or an adhesive agent.

Next, a detailed description will be given of the structure for joining the rotary shaft **41** and the hub **42** of the spindle motor in accordance with this embodiment. As shown in FIG. **2**, the joining length of the rotary shaft **41** and the hub **42** is longer than the joining length of the rotary shaft **41** and the thrust plate **43**, but if the overall height of the motor is shortened, the joining length of the rotary shaft **41** and the hub **42** also inevitably becomes short. Consequently, since the joining strength of the rotary shaft **41** and the hub **42** declines. Accordingly, in this embodiment, in the same way as the structure for joining the rotary shaft **41** and the thrust plate **43**, the two members are joined by welding after the rotary shaft **41** and the hub **42** are press-fitted with an appropriate press-fitting force.

Here, if press-fitting is effected by providing a large press-fitting allowance of the hub **42** with respect to the rotary shaft **41**, distortion occurs in the hub **42** due to the press-fitting stress. Consequently, the perpendicularity of the hub **42** with respect to the rotary shaft **41**, specifically the perpendicularity of the disk-mounting surface **42***b* of the hub **42** with respect to the rotary shaft **41**, becomes deteriorated, so that the problem of occurrence of runout exceeding an allowable range is liable to occur when the disk is mounted on the hub **41** and is rotatively driven.

Accordingly, in this embodiment, an axially recessed relief portion **50** is formed at the joining interface between the rotary shaft **41** and the hub **42**, and the two members are joined by laser welding in this relief portion **50**. The relief portion **50** is formed by a tapered surface **41***b* formed at a corner of the tip of the rotary shaft **41** and a tapered surface **42***c* formed at an inner peripheral edge of a shaft-attaching hole **28** of the hub **42**. Of these tapered surfaces, the tapered surface **41***b* of the rotary shaft **41** also functions as a guide portion at the time of press-fitting the hub **42** to the rotary shaft **41**. It should be noted that, in this embodiment, since a damper guide **29** for guiding a damper (not shown) for

holding the disk is provided on an upper end face of the hub **42** in such a manner as to axially project slightly from the joining interface between the rotary shaft **41** and the hub **42**, the attempt to make the motor thin is not hampered even if the relief portion **50** is not formed. Further, as for the welding position, the entire periphery of the joining interface may be welded, or welding may be effected partially at a plurality of locations.

By virtue of the above-described joining structure, since the joining strength of the rotary shaft **41** and the hub **42** can be sufficiently increased without forcibly press-fitting the rotary shaft **41** and the hub **42**, the shock resistance of the motor improves, and the perpendicularity of the disk mounting surface **42***b* of the hub **42** with respect to the rotary shaft **41** can be maintained with high accuracy.

FIG. **5** is a half cross-sectional view showing a spindle motor in accordance with a fourth embodiment of the invention. In FIG. **5**, those arrangements having common functions to those of the spindle motor shown in FIG. **2** are denoted by the same reference numerals, and a detailed description thereof will be omitted.

The stator cores **34** each having the coil **35** wound therearound are attached to the outer periphery of a tubular holder **32'** provided uprightly in the center of the fixing frame **31**. This tubular holder **32'** is formed to be axially longer than the tubular holder **32** shown in FIG. **2**, and the bearing sleeve **33** and the counter plate **44** are fixed to its inner periphery. Namely, although the counter plate **44** in FIG. **2** is joined to the opening of the bearing sleeve **33**, in FIG. **5**, the counter plate **44** is joined to the opening of the tubular holder **32'** of the fixing frame **31** after being press-fitted thereto with an appropriate press-fitting force.

In joining the counter plate **44** to the tubular holder **32'**, the axially recessed relief portion **60** is provided at the joining interface between the two members, and the counter plate **44** and the tubular holder **32'** are welded in this relief portion **60** to integrate the two members. As the welding process, the arc welding process or the electron beam welding process is adopted as described above. Preferably, however, at least one of the counter plate **44** and the tubular holder **32'** is fused by the electron beam welding process typified by laser welding so as to join the two members. By joining the two members in the relief portion **60** in this manner, since a portion projecting from the bottom surface of the fixing frame **31** or the counter plate **44** is not formed, the attempt to make the motor thin is not hampered. Further, since the fixed shaft **31** and the counter plate **44** are firmly joined by welding, the shock resistance also improves.

In this embodiment as well, the rotary shaft **41** and the thrust plate **43** are joined in the same way as in the above-described embodiments. Namely, one end of the rotary shaft **41** is press-fitted in the central hole of the thrust plate **43**, the relief portion **70** is formed at the joining interface between the rotary shaft **41** and the thrust plate **43**, and the two members are integrated by welding in the relief portion **70**.

Further, in the joining of the rotary shaft **41** and the hub **42**, in the same way as the joining of the rotary shaft **41** and the thrust plate **43**, the rotary shaft **41** is press-fitted in the central hole of the hub **42**, the relief portion **50** is formed at the joining interface between the rotary shaft **41** and the hub **42**, and the two members are integrated by welding in the relief portion **50**. Incidentally, this relief portion **50** may be omitted depending on the shape of the hub **42**.

As described above, in accordance with the spindle motor shown in FIG. **5** as well, it is possible to obtain a sufficient joining strength even if the joining length of the rotary shaft **41** and the thrust plate **43** and the joining length of the tubular holder **32'** of the fixing frame **31** and the counter plate **44** are relatively short. Accordingly, it is possible to stably maintain the perpendicularity of the thrust plate **43** with respect to the rotary shaft **41**. Moreover, even if projections are formed by welding, since the projections are respectively accommodated in the relief portions **60** and **70**, the attempt to make the overall motor thin is not hampered. Further, since the rotary shaft **41** and the thrust plate **43** are joined by welding, even if the lubricating fluid **5** is oil, catalytic action does not occur, and the characteristics of the lubricating fluid **5** do not deteriorate.

Next, a description will be given of the structure for joining the fixed shaft **16** and the thrust plate **23** in FIG. **1**. After the bearing sleeve **21** formed integrally with the hub **22** is fitted over the fixed shaft **16** provided uprightly on the fixing frame **11**, the annular thrust plate **23** is press-fitted to the fixed shaft **16** with an appropriate press-fitting force. Subsequently, as the joining interface portions of the fixed shaft **16** and the thrust plate **23** are welded together, the two members are joined. As shown in FIG. **6**, the relief portion **70** which is recessed below the hydrodynamic surface is annularly formed at the peripheral edge of the central hole corresponding to the joining interface portion on the thrust plate **23** side. The laser welding process is desirable as this welding, and the thrust plate **23** formed of a copper group material, a stainless steel metal, or the like is fused so as to undergo metallic fusion with the fixed shaft **16**. The welding with the fixed shaft **16** is performed in the relief portion **70**, and the arrangement provided is such that even if a local projection occurs due to welding, it does not project above the hydrodynamic surface.

By virtue of such an arrangement, since a sufficient joining strength can be obtained even if the joining length of the fixed shaft **16** and the thrust plate **23** is relatively short, the perpendicularity of the thrust plate **23** with respect to the fixed shaft **16** can be maintained stably, so that the reliability of the motor improves. Moreover, since the axially recessed relief portion **70** is provided at the joining interface, and the two members are integrated by welding in this relief portion **70**, the attempt to make the overall motor thin is not hampered. Further, since the joining interface portions located in such a manner as to be contiguous to the lubricating fluid **5** for generating hydrodynamic pressure are welded, even if the lubricating fluid **5** is oil, catalytic action does not occur, and the characteristics of the lubricating fluid **5** do not deteriorate.

Although a description has been given above specifically of the embodiments of the invention devised by the present inventors, the invention is not limited by the foregoing embodiments, and it goes without saying that various modifications are possible without departing from the scope of the invention.

For example, although, in the above-described embodiment, an example has been shown in which joining is accomplished by welding in such a way that the counter plate **44** closes the bearing sleeve **33** or the opening of the tubular holder **32'** of the fixing frame **31**, part of the joining interface may be welded to secure joining strength, and the entire periphery of the joining interface may be sealed by an adhesive agent. Consequently, it is possible to reliably prevent the leakage of the lubricating fluid.

Furthermore, the invention is similarly applicable to a spindle motor than a hard-disk driving motor, e.g., a CD-ROM driving motor and a polygon-mirror driving motor.

What is claimed is:

1. A spindle motor comprising:

a fixed shaft;

a cylindrical rotary bearing member rotatably supported on an outer peripheral face of the fixed shaft, and made of a first metal material;

a rotary hub integrally joined to the rotary bearing member, and made of a second metal material different from the first metal material; and

a potential-difference alleviating member provided on the joining surfaces of the rotary bearing member and the rotary hub, and made of a third metal material whose ionization tendency in an electrochemical series is positioned between ionization tendencies of the first and second metal materials.

2. The spindle motor as set forth in claim **1**, wherein the first metal material is a copper group metal material, the second metal material is an aluminum group metal material, and the third metal material is a nickel group metal material.

3. The spindle motor as set forth in claim **1**, wherein the potential-difference alleviating member is formed on at least one of the joining surfaces of the rotary bearing member and the rotary hub by any one of plating, vapor deposition and coating.

4. A spindle motor comprising:

a fixed shaft;

a cylindrical rotary bearing member rotatably supported on an outer peripheral face of the fixed shaft, and made of a first metal material; and

a rotary hub integrally joined to the rotary bearing member, and made of a second metal material different from the first metal material; and

a passivation film formed on the joining surfaces of the rotary bearing member and the rotary hub.

5. The spindle motor as set forth in claim **4**, wherein the passivation film is made of either the first metal material or the second metal material.

6. The spindle motor as set forth in claim **5**, wherein the passivation film is made of a third metal material which is different from the first and second metal materials.

7. A spindle motor comprising:

a fixed frame made of a first metal material;

a cylindrical fixed bearing member integrally joined to the fixed frame, and made of a second metal material different from the first metal material;

a rotary shaft rotatably supported on an inner peripheral face of the fixed bearing member;

a rotary hub secured to the rotary shaft; and

a potential-difference alleviating member provided on the joining surfaces of the fixed frame and the fixed bearing member, and made of a third metal material whose ionization tendency in an electrochemical series is positioned between ionization tendencies of the first and second metal materials.

8. The spindle motor as set forth in claim **7**, wherein the first metal material is a copper group metal material, the

second metal material is an aluminum group metal material, and the third metal material is a nickel group metal material.

9. The spindle motor as set forth in claim **7**, wherein the potential-difference alleviating member is formed on at least one of the joining surfaces of the rotary bearing member and the rotary hub by any one of plating, vapor deposition and coating.

10. A spindle motor comprising:

a fixed frame made of a first metal material;

a cylindrical fixed bearing member integrally joined to the fixed frame, and a second metal material different from the first metal material;

a rotary shaft rotatably supported on an inner peripheral face of the fixed bearing member;

a rotary hub secured to the rotary shaft; and

a passivation film formed on the joining surfaces of the fixed frame and the fixed bearing member.

11. The spindle motor as set forth in claim **10**, wherein the passivation film is made of either the first metal material or the second metal material.

12. The spindle motor as set forth in claim **11**, wherein the passivation film is made of a third metal material which is different from the first and second metal materials.

13. A spindle motor comprising:

a fixed shaft;

a cylindrical rotary bearing member rotatably supported on an outer peripheral face of the fixed shaft, and made of a first metal material;

a rotary hub integrally joined to the rotary bearing member, and made of a second metal material different from the first metal material; and

an insulating resin film formed on the joining surfaces of the rotary bearing member and the rotary hub.

14. The spindle motor as set forth in claim **13**, wherein the resin film is formed on outer circumferential faces of the rotary bearing member and the rotary hub continuously from the joining surfaces such that the rotary bearing member and the rotary hub are partly conducted.

15. A spindle motor comprising:

a fixed frame made of a first metal material;

a cylindrical fixed bearing member integrally joined to the fixed frame, and made of a second metal material different from the first metal material;

a rotary shaft rotatably supported on an inner peripheral face of the fixed bearing member;

a rotary hub secured to the rotary shaft; and

an insulating resin film formed on the joining surfaces of the rotary bearing member and the rotary hub.

16. The spindle motor as set forth in claim **15**, wherein the resin film is formed on outer circumferential faces of the rotary bearing member and the rotary hub continuously from the joining surfaces such that the rotary bearing member and the rotary hub are partly conducted.

* * * * *